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Programme des pompes de chaleur

Simulation of an Industrial Heat Pump

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Simulation of an Industrial Heat Pump

NATIONAL RESEARCH COUNCIL
Energy Division
Heat Pump Section

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SUMMARY

Presented is a proposed simulation for a vapour compression heat pump which supports a steam generating capacity. Its general approach to simulation can be exploited as an optimization tool allowing the best combination of environment conditions and size to be determined for a given application.

The report details the developmental considerations during simulation synthesis. It presents the various equations which constitute the heat pump components. The relationships are presented in detail and the logic flowcharted for ease of programming. A discussion of limitations to the system is presented and recommendations are made.

It was concluded that the simulation is incomplete. In its current form, the simulation has been untested. It is recommended that validation proceed on the basis of comparison with real data from industrial applications as a next phase in the simulation's development.

RÉSUMÉ

Le présent rapport expose la simulation proposée d'une pompe à chaleur à compression de vapeur associée à un générateur de vapeur. La démarche générale adoptée pour la simulation peut être exploitée comme une méthode d'optimisation permettant de déterminer la meilleure taille en fonction des conditions environnementales, pour une application donnée.

Le rapport décrit en détail l'élaboration du programme de simulation. Il présente les diverses équations représentant les éléments de la pompe à chaleur. Les relations sont présentées de façon détaillée, avec un organigramme pour faciliter la programmation. Les limites du système sont examinées et des recommandations sont présentées.

On a conclu que la simulation était incomplète. Elle n'a pas été essayée dans sa forme actuelle. Il est recommandé que l'étape suivante consiste à valider le programme de simulation en comparant ses résultats avec des données réelles obtenues dans l'industrie.

1.0 INTRODUCTION

This report presents a proposal for steady state simulation of a vapour compression heat pump for use on a micro computer. The primary intent of this simulation is to reflect the performance of different heat pumps in sufficient detail to allow for comparison. In this manner, by varying loads or capacities as well as environment conditions, a series of simulations can provide an optimum configuration. Although a model of this type is ideally suited for incorporation within the framework of a larger system, its structure is self-contained and can be easily integrated within other systems.

One characteristic of this simulation is its generality and wide range of applicability. Provided major component compatibility exists, a simulation of this nature can afford to ignore minor discrepencies. This model is not concerned with how precisely a heat pump can be modelled insomuch as whether it can detect macroscopic performance tendencies between different units. This is accomplished by extracting the major components into a rudimentary "generic" model and performing a detailed analysis on it. Because the measure of performance is system COP (coefficient of performance), an analysis of major componenets in the generic model adequately furnishes an acceptable index of performance.

The effectiveness of the simulation increases as a heat pump's resemblance to the generic unit improves. Essentially, the generic heat pump can be described as a hot water or steam generating heat pump with these major horizontally oriented components: compressor, tube-and-shell condenser and tube-and-shell flooded evaporator and expansion device. The refrigerant completes a unitary cycle through the shellside of each heat exchanger. Both fundamental heat transfer and thermodynamic relationships as well as published component performance data are utilized to this end.

The simulation requires inputs of two natures. First, physical data must be supplied to adequately describe physical setup and dimensions of components. Second, variable data completely mapping the thermodynamic property range of the refrigerant and coolant, including data mapping the operation of the components modelled with performance specifications, is furnished.

Essentially, the report accomplishes two objectives. It presents the various relationships describing the operation of the heat pump and provides a methodology for their solution. Equations are presented for each significant heat pump component. Together, they form a system of equations requiring simultaneous solution, the nature of which becomes iterative. Many techniques are available for use. A reliable and quick method is presented in this report. However, since this report has not preoccupied itself with optimizing a selection of non-linear system solvers, it is highly likely that other equally effective or superior techniques exist. Subsequently, the equations are presented in a format facilitating submissions to such alternate approaches.

The model is presented in two styles. Section Four presents the various relationships pertinent to each component in text book fashion. This facilitates initial reading and evaluation. Section Five presents the same material in a format advantageous for submission to simultaneous non-linear solvers. A close inspection reveals consistent variables and units (imperial), property data properly referrenced to specific conditions, special function delimiters, variable sheets. In summary, Section Five is intended for the computer programmer possessing limited practical knowledge of the material. The order of presentation is logical and permits top-down design. Furthermore, flow charting is provided. Programming efficiency is not one of the purposes of this section. In fact, for clarity's sake, equations are explicity presented at the expense of efficiency. This is left as a task for the programmer.

In the end, this report amounts to a first draft of the proposed simulation system. Indeed, refinements can be made. For example,

assumptions may prove invalid and revision became necessary. In addition, certain correlations may be improved by researching already published material or using that yet to appear. It is most important to realize that this system has not yet been tested as an integrated whole.

2.0 CRITERIA AND CONSTRAINTS

Certain criteria can be identified in controlling the environment of the simulation. The most obvious involves the purpose of entire exercise. Basically, a simulation system is required which will enable heat pumps to be compared on the basis of overall performance such as COP. Parameters allowing for optimization are the source and sink environment as well as the overall load or capacity being considered.

Another concern is the ability to insure verification. The simulation is meaningless unless its output can be validated by real heat pump data. Therefore, it should remain compatible with units scheduled for data acquisition tests. Subsequently, an iterative system improvement can then be established reciprocating between comparison and modification. Effectively, effort should be concentrated on industrial type applications.

Currently, this simulation is envisioned to reside on a micro computer. Subsequently, its design should recognize the micro computer environment.

Moreover, an implicit requirement involves the awareness that the system is subject to real world data availability and reliability. Hense, practicality is important.

3.0 DEVELOPMENT CONSIDERATIONS

This section is dedicated to the presentation and discussion of the developmental factors governing the design of the simulation. The aspects discussed in this section can be categorized under hardware, efficiency and practicality consideraions.

3.1 HEAT PUMP CONFIGURATION

The most common industrial heat pump cycle is the vapour compression cycle. By designing the simulation on the basis of existing applications, the validity of the simulation can be verified by comparison of predicted performance with actual figures. To this extent, it was of vital importance that the heat pump configuration chosen be compatible with existing units labelled for data acquisition purposes.

Appendix (B) contains specifications for a heat pump typically satisfying these requirements. Briefly, these are halogen refrigerant, evaporator, condenser/steam generator, liquid subcooler, suction/liquid heat exchanger, high pressure receiver, oil system, compressor, motor, control system, see figure (1). However, only a portion of these are thermodynamically significant. The relevant information can be extracted:

flooded evaporator
hot water/steam generating condenser
liquid subcooler
screw compressor

Furthermore, it is implicitly discerned that an expansion device exists and that the cycle is unitary. Bleedflows can be disregarded for simplicity without excessive loss in validity. The result can be

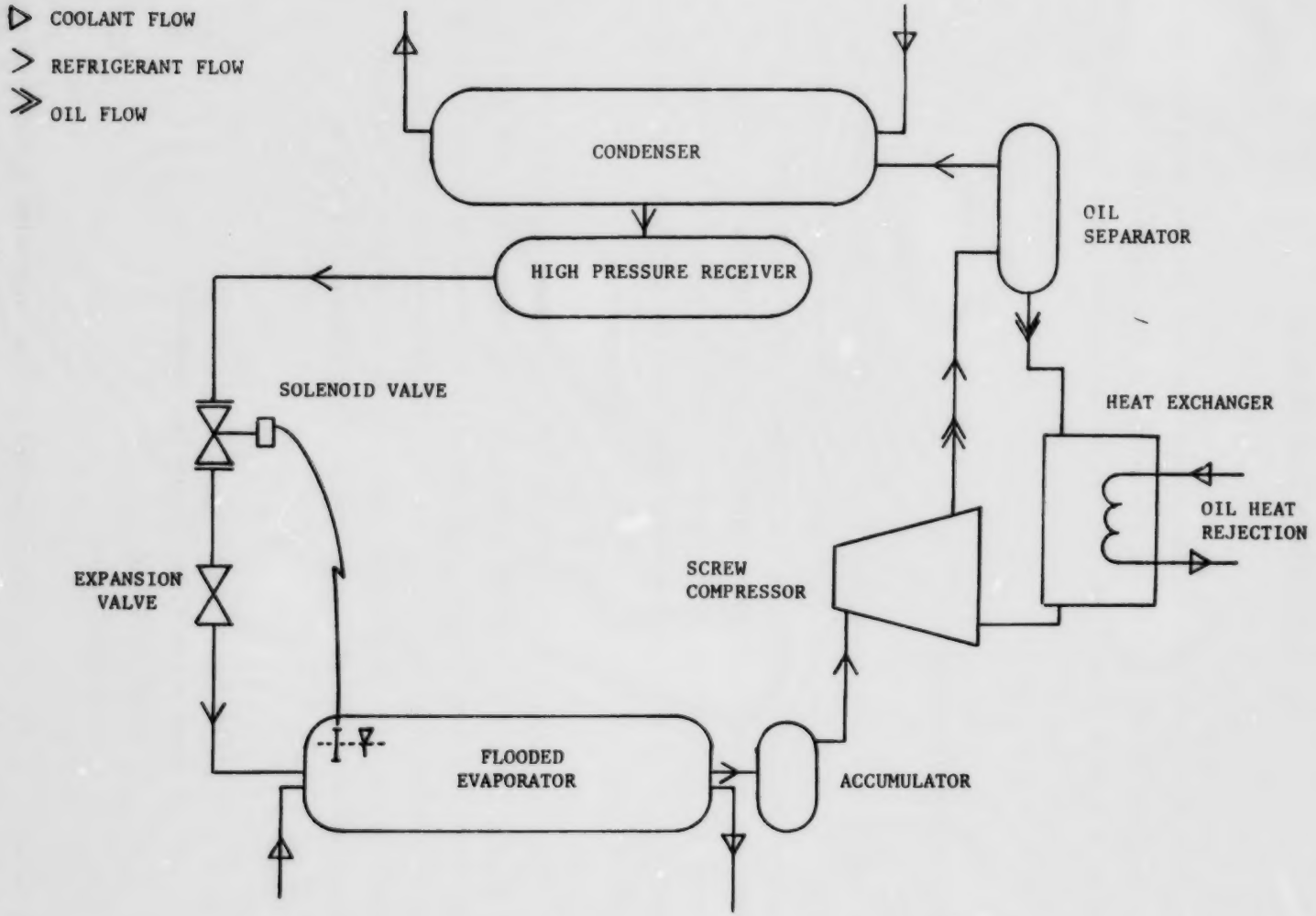

Fig (1): Simplified schematic of a real heat pump cycle

referred to as a "generic" heat pump. Although physically unrealistic, modelling its components will provide a reasonable estimate of performance in a unitary cycle. By omitting motor energy consumption and focussing strictly on brake horsepower, complications involving motor fuel type can be avoided. Instead, efficiency discrepencies between electric, gas or oil driven compressor motors can be accounted by a nominal efficiency factor when determing the end coefficient of performance (COP). Because pumps consume only a negligible fraction of the compressor power, their contributions can be ignored.

This approach provides many advantages. It achieves a markedly less complex model at an affordable loss in precision. The objective of this simulation is its ability to predict and compare macro performance between heat pumps. This is achieved by concentration on the major components. Furthermore, its generality insures compatibility with the maximum number of heat pumps as possible. In fact, this package can even be useful in modelling more complex systems such as multi-stage units in that it can provide an estimate of the lower bounds of the more effective and efficient heat pump.

3.2 MODELLING APPROACH

This section discusses the selection either published performance data or fundamental heat transfer approach in modelling each component.

Generally in the case of heat exchangers a case can be made for adopting the fundamental approach to modelling. The major concern is the availability of performance data which often does not exist. Although firms may manufacture heat exchangers, they may only do so on an order basis and not actually stock off-the-shelf type units. In other cases, this data may only reside in design programs used by the engineering department of such firms. In any case, availability cannot be guaranteed. Another consideration involves the homogeneity of specifications which are published. Because of inconsistency in the presentation of data, system design becomes difficult and may involve a great degree of alteration before it can be used. Finally, the reliability of available data is an unknown factor. Data integrity, in view of advertising and competitive considerations, is questionable. Also, there is a complete lack of knowledge concerning the controls and criteria of the testing procedure surrounding that information.

Conversely, performance data is used to model the screw compressor. the reasoning here is twofold. Most important, it is extremely difficult to accurately model compressors using a heat transfer approach. In addition, available compressor performance data is complete and meticulous and hence leaves little doubt as to reliablity. Appendix (C) shows typical compressor performance data.

3.3 SIMULATION THEORY

Simulation refers to the successful identification of a physically attainable thermodynamic refrigerant cycle for a specified hardware configuration. This section will discuss how this requirement is interpretted in this model.

Consider the "black box" schematic representation of a heat pump as shown in fig. (2).

A thermodynamic analysis provides 3 equations, namely,

$$\text{(I)} \quad Q_{in} + P = Q_{out} + OHR$$
$$\text{(II)} \quad Q_{in} = M_{SRC}\, C_p\, (T_1 - T_2)$$
$$\text{(III)} \quad Q_{out} = M_{SK}\, C_p\, (T_8 - T_7)$$

Because there are more unknowns than equations, this system cannot be solved. Additional equations are obtained from the modelling of the interior of the heat pump. However, these relations are dependent upon the internal operating cycle which is as yet unknown. Consequently, an iterative scheme is employed where initial guesses are furnished. Validity of these guesses is determined by the satisfaction of the following functions:

$$\text{(IV)} \quad M_R = M_X$$
$$\text{(V)} \quad M_R\, \Delta h_c = Q_c$$
$$\text{(VI)} \quad M_R\, \Delta h_E = Q_E$$
$$\text{(VII)} \quad Q_c = Q_{out}$$
$$\text{(VIII)} \quad Q_E = Q_{in}$$

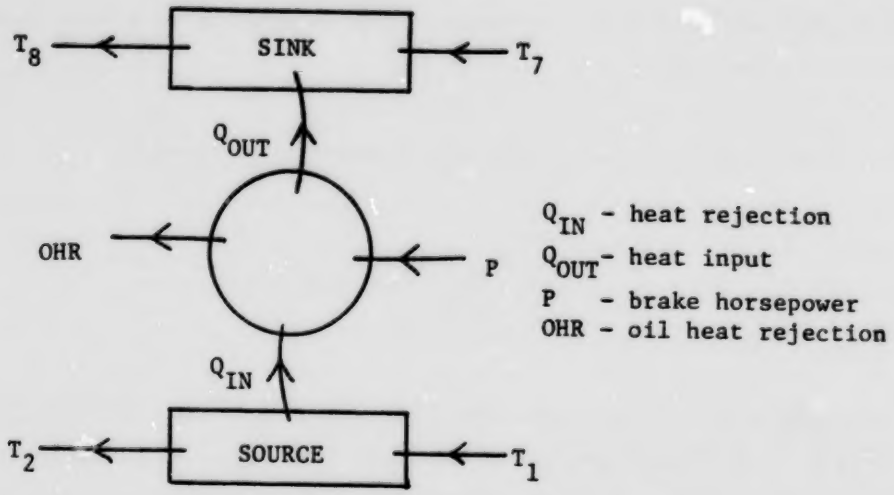

Fig (2): Overall heat pump thermodynamic analysis

where M_R = refrigerant flow rate from compressor model
M_X = refrigerant flow rate from expansion device model
Δh_C = enthalpy change across condenser
Δh_E = enthalpy change across evaporator
Q_C = condenser model heat transfer
Q_E = evaporator model heat transfer

Equation (IV) represents a mass balance while the remaining equations involve energy balances. Equations (V) and (VI) insure that the theormodynamic heat transfer matches the calculated heat transfer from the heat exchanger model. Hence, theoretical expectation must agree with physical reality. Equations (VII) and (VIII) establish the relationship with the environmental conditions underwhich the heat pump operates.

The models of the various heat pump components can be viewed as dependent upon eight variable conditions:

(i) ET = nominal evaporating temperature
(ii) CT = nominal condensing temperature
(iii) ΔT_{SH} = suction superheat
(iv) ΔT_{SC} = liquid subcool
(v) M_{SK} = sink fluid flowr rate
(vi) M_{SRC} = source fluid fow rate
(vii) T_2 = source fluid outlet temperature
(viii) T_8 = sink fluid outlet temperature

These are the variables whose values must be initially estimated. In addition, two unknowns which must be provided are.

T_1 = source fluid inlet temperature
T_7 = sink fluid inlet temperature

In fact, the statepoints must be clearly defined at 1 and 7 since reference pressures are also required. These ten variables completely define the operation of a given cycle. Since variables (i) to (viii) are repeatedly estimated, a mathematical scheme is used to predict the next best guesses. A solution is obtained when a given vector solution satisfies the eight equations within a specified tolerance. Figure (3) shows a schematic representation of the iteration process.

The major amount of heat transport occurring in a vapour compression heat pump is accomplished through the phase transformation of the refrigerant. The superheat stage of the cycle contributes only fractionally to the quantity of heat transferred. Consequently, the system solution is only weakly dependent on changes in ΔT_{SH}. Conceivably, solution convergence could occur even with unrealistic estimates of ΔT_{SH}. Therefore, it is suggested that reasonable initial estimate be furnished. As a result, convergence on nearby and more likely values would be expected.

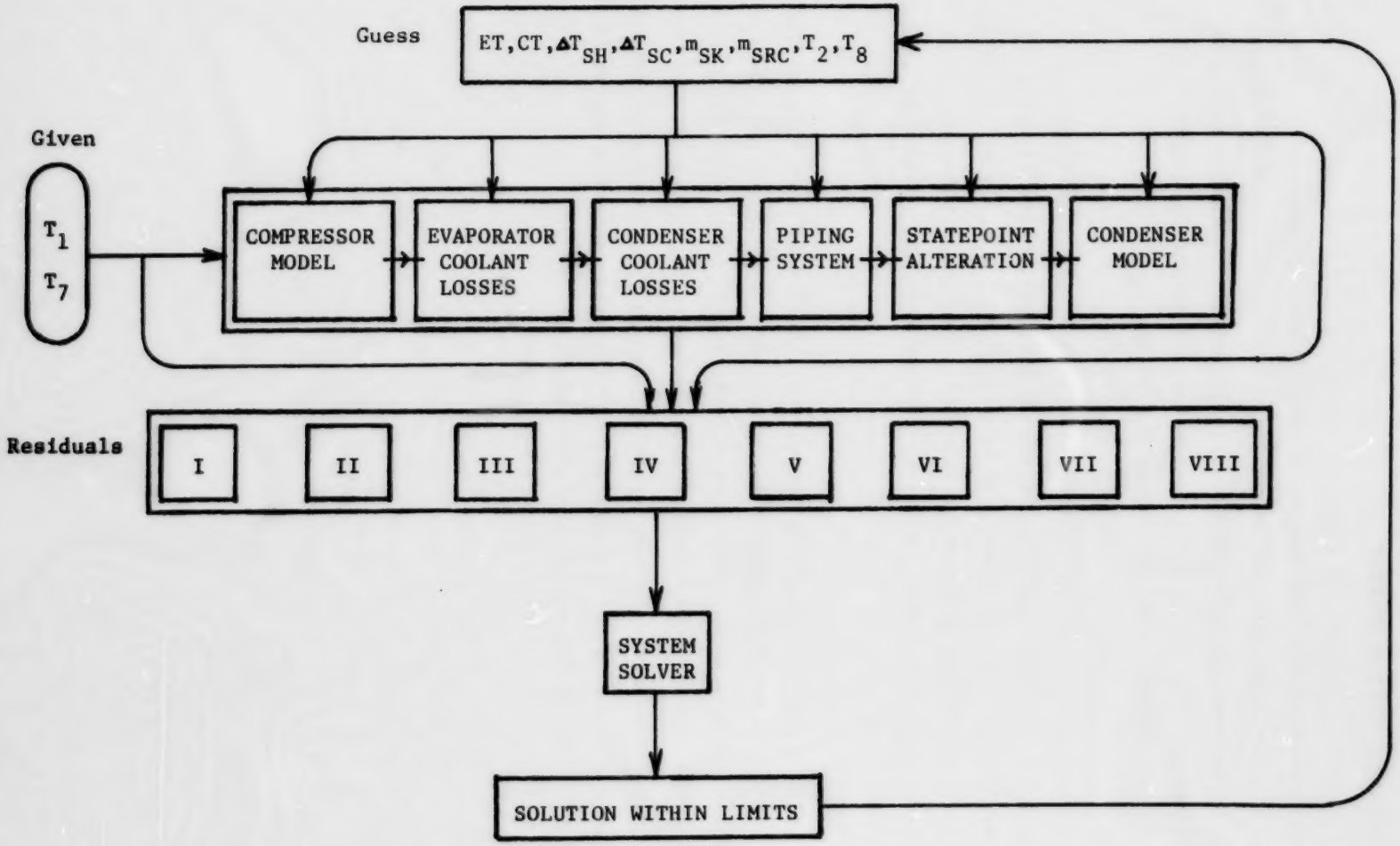

Fig (3): Schematic representation of the iteration process

3.4 NON-LINEAR SYSTEM SOLVERS

Heat pump operation is inherently non-linear. Consequently, a convergence technique for a non-linear system of simultaneous equations is recommended in order to effectively provide iterative solution estimates. The two factors to consider in selecting a method are speed and the likelihood of divergence occurring.

Higher order methods are quicker and less likely to diverge. An example of a first-order technique is the Newton-Raphson Method. Brown's method is an example of a second-order technique. It is much quicker and possesses a wider range of convergence than the Newton-Raphson melthod. A Basic version of Brown's Method is included in Appendix (E).

Another method of potential interest is the Nedler-Mead Simplex Solver. This is another higher order technique which boasts the advantage of allowing the solution domain to be restricted. This approach may be required should Brown's Method prove prone to wandering into unmapped a data regions even with correct relationships describing the system. This might occur, for example, if the iterative solution estimates were widely oscillatory during the convergence procedure.

3.5 TRANSIENCY AND STEADY-STATE

According to Al-Juwayhel [8], heat pump capacities can be represented by the first order response,

(1) $$\frac{Q_{CAP}}{Q_{SS}} = 1 - e^{-t/\tau}$$

where Q_{CAP} = heat pump cycle capacity
Q_{SS} = heat pump steady-state capacity
t = time
τ = heat pump time constant

Solving the differential equation yields,

(2) $$Q_{CAP} = Q_{SS}\ (t_o - \tau(1-e^{-to/\tau}))$$

where t_o = on-time

Hence, it is seen that for large on-time, Q_{CAP} approaches Q_{SS}. This simulation is concerned with industrial application. Since down-times are not forseen and because most industrial heat pumps are designed for continuous operation, Q_{CAP} effectively equals Q_{SS}. Consequently, the added difficulty in modelling, by determining the time constant, is not warranted.

3.6 INPUT/OUTPUT OPERATIONS

The nature of the simulation being considered necessarily retrieves refrigerant, coolant property data, and component hardware specifications frequently. Because such a quantity of data may not be able to reside entirely on a micro-computer's memory due to size constraints, data must be obtained from files. Since, input/output (I/O) operations are extremely time consuming on any computer system, long execution periods may result if many individual heat pumps are to be simulated. Ordinarily, in a large computer system, data is

initially read from files into arrays and stored there for remainder of the simulation. This reduces execution time.

Usually, data exists as discrete sets of values which describe a n-dimensional surface. Table reading procedures will delay the lookup procedure if interpolation is performed between points. Delays in interpolation and I/O operations can be reduced by using data fits wherever possible and storing correlation coefficients instead. Since each correlation substitutes for an indefinite number of discrete points, this substantially reduces I/O operation. In fact, further savings can be incurred if the coefficients themselves can be stores in direct memory.

Generally, it was found that compressor performance data lends itself easily to curve fitting* and this procedure is recommended. On the contrary, thermodynamic properties are more difficult to correlate within acceptable limits of less than one percent using common linearized standard curves (See Ref. [11]). Subsequently, prioritizing compressor data over thermodynamic in cases where limited memory is available for data storage is recommended.

Note that some correlations for refrigerant data do exist (Appendix A) providing excellent agreement with real data. In the future, further correlations may be published, therefore, this system should be appropriately updated upon that eventuality.

---

* Some data was found to fit the following standard equation,

$Y = A_0 + A_1 X$, where $Y = \log y$, $X = \log x$

Occassionally a third term may be necessary ($A_2 X^2$) or variation thereof.

4.0 COMPONENT MODELLING

This section provides a detailed look at the models used to simulate the operation of the heat pump components. There are five components considered in accordance with the "generic" heat pump model: compressor, condenser, evaporator, expansion device and piping. Three of these, (condenser, evaporator and piping) are fundamental in approach utilizing heat transfer and thermodynamic principles. The condenser and expansion device are modelled using published performance data. In addition, another module, the "statepoint alterations" module uses pressure drop results and alters the cycle as identified by the compressor model. Although presented as a seperate module, it does not exist as a distinct component itself. Figure (4) shows the generic heat pump being modelled with all statepoints identified. Figure (5) and Figure (6) display the refrigerant and coolant thermodynamic cycles, respectively.

4.1 COMPRESSOR

Due to the difficulty in modelling a compressor using fundamental relationships, manufacturer published performance specifications are used instead.

By observing available screw compressor data, an evaluation concerning that which can be considered reasonable expectations for compressor data was made. Data selected for compressor modelling should provide brake horsepower, oil heat rejection rate, refrigerant mass flow, discharge temperature and refrigeration load as a function of evaporation and condensation temperatures. Generally, compressor performance tables specify figures based on a constant superheat and subcool. For values different than those specified, the capacity or refrigeration load will vary. the following are corrections to be applied to the quoted capacity value. For a subcooling change,

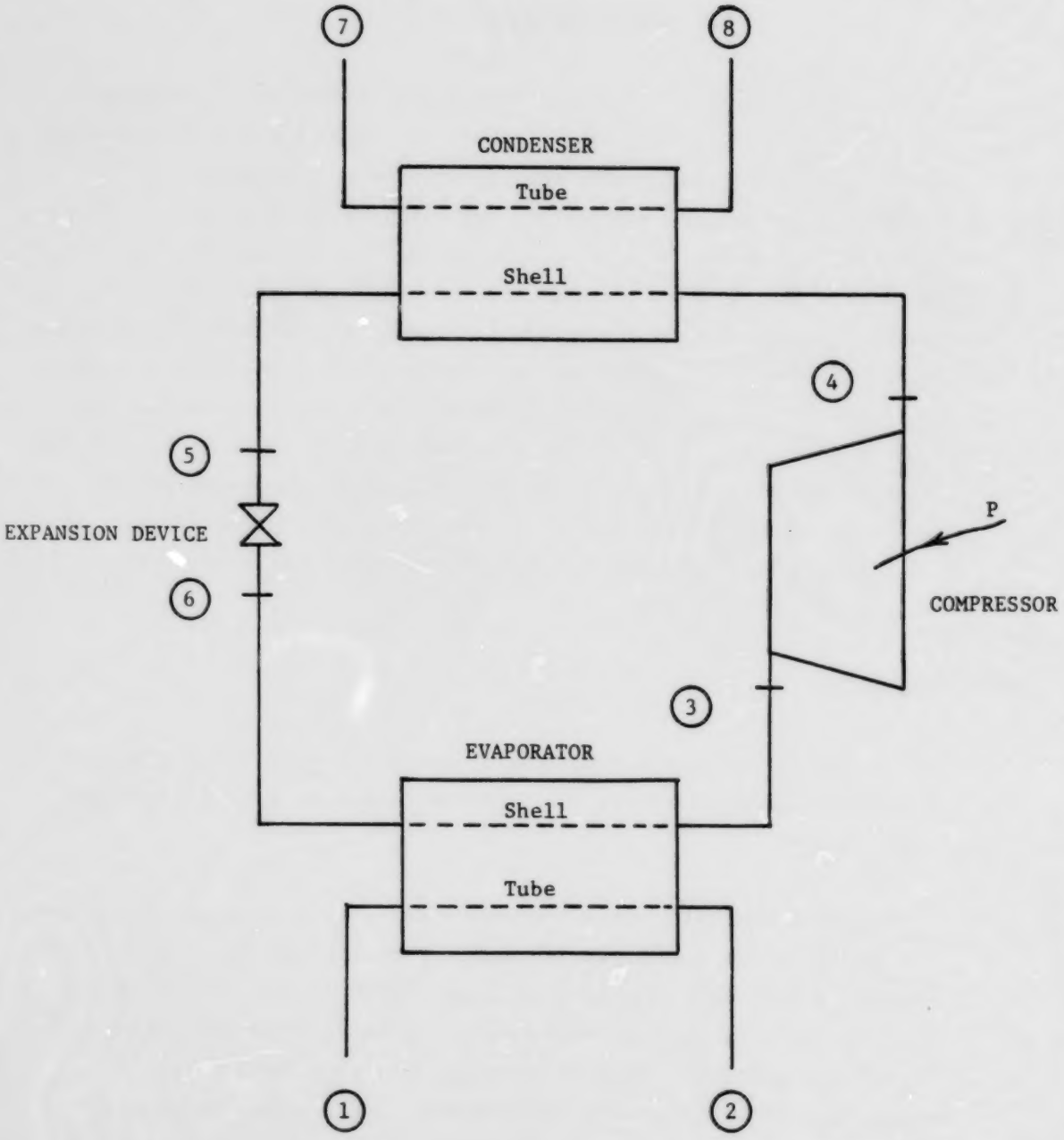

Fig (4): The "Generic" heat pump with statepoints identified

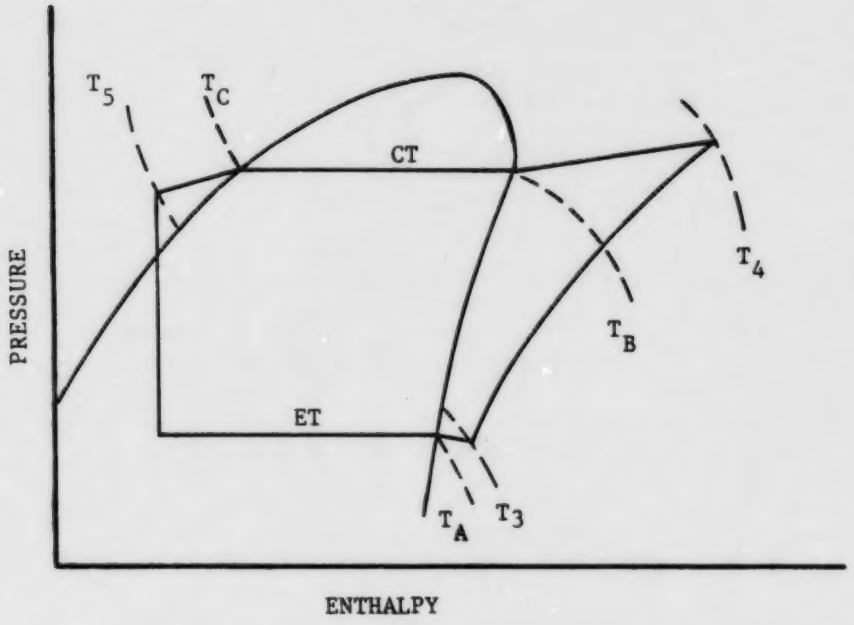

Fig (5): Refrigerant thermodynamic cycle with pressure losses

——— steam generating condenser flow
—·— hot water generating condenser flow
– – – – evaporator flow

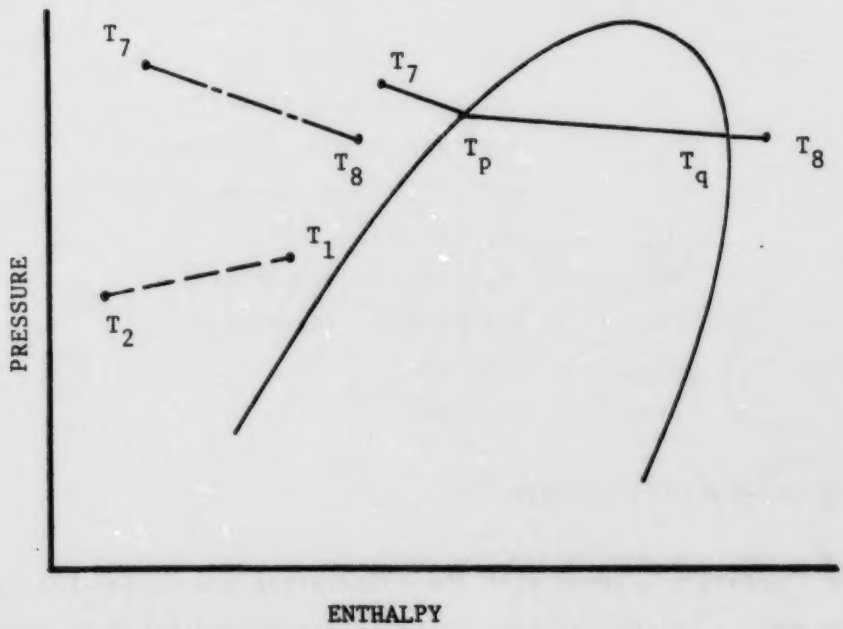

Fig (6): Coolant thermodynamic cycle with pressure losses

(3) $$Q_{CORR} = M_R \, C_{PR\ell} \, \Delta T_{CORR}$$

while with respect to superheating,

(4) $$Q_{CORR} = M_R \, C_{PRv} \, \Delta T_{CORR}$$

where Q_{CORR} = capacity correction

M_R = refrigirant mass flow rate

$C_{PR\ell}$ = liquid refrigerant specific heat

C_{PRv} = vapour refrigerant specific heat

ΔT_{CORR} = positive or negative change in the amount of superheat or subcool

Symbollically, the results of this data search can be shown as,

$$(P, OHR, M_R, T_4, Q_{in}) = f\,(ET, CT, \Delta T_{SH}, \Delta T_{SC})$$

where
- P = brake horsepower
- OHR = oil system heat rejection rate
- M_R = refrigerant mass flow rate
- T_4 = vapor discharge temperature
- Q_{in} = refrigeration load
- ET = evaporation temperature
- CT = condensation temperature
- ΔT_{SH} = superheat
- ΔT_{SC} = subcool

4.2 SHELL-AND-TUBE HEAT EXCHANGERS

This simulation assumes that the evaporating and condensing subcooling requirements of the refrigerant cycle are accomplished by horizontal

shell-and-tube heat exchangers with shellside refrigerant flows. Heat exchanger analysis is based on the effectiveness-NTU method. To apply this approach certain simplifications of heat exchanger operation were adopted.

Since heat exchangers exist in widely varying permutations of coolant and shell passes, shellside temperature gradients are bound to exist. Modelling for these variations is extremely difficult. To avoid this problem, it is assumed that the shellside vapour is well-mixed and homogeneous. This is, in fact, the case in the evaporator which contains only saturated liquid.

The two-phase character of the fluid flow complicates the effectiveness approach to modelling the heat exchangers. The difficulty is that effectiveness method utilizes hot and cold fluid heat capacity rates ($M C_p$). The rates change abruptly from a finite value during single phase state to an infinite one during a two-phase regime. The following simplification is valid for heat exchangers with single phase coolant flow in the tubes. The steam generating condenser will be mentioned separately in Section (4.2.2).

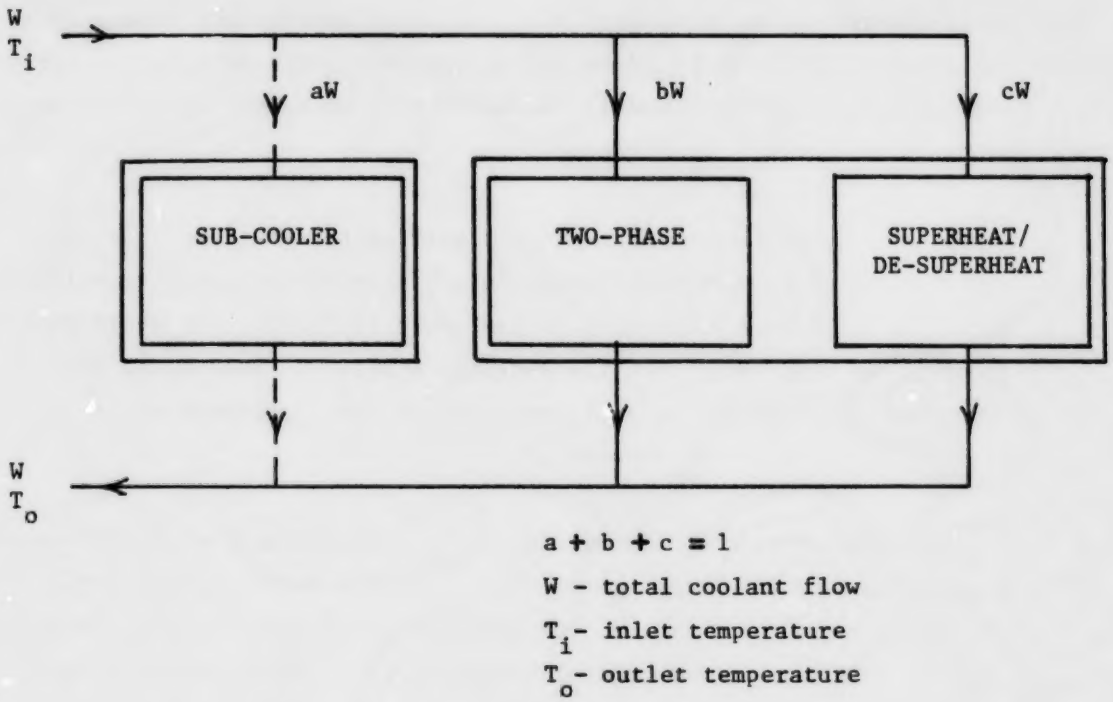

Fig (7): Separation of heat exchanger functions

Figure (7) is a schematic representation of the segregation of heat exchanger functions. The dashed subcooling function reflects that the operation is restricted only to the condenser* side of the heat pump.

To demonstrate the simplification, consider the condenser. Assuming a homogeneous shellside temperature distribution, de-superheating will occur on the tubes throughout the cavity. Hence, no specific length of tubing is responsible for de-superheating. Instead, de-superheating can be envisioned as the heating of a portion of the total water flow (cW) from T_i to T_f. Similarly, the entire length of tubing contributes to condensation of the saturated refrigerant vapour. Accordingly, the heating of bW from T_i to T_f is attributed to the heat transfer derived from the latent heat of refrigerant. The subcooler part is analysed in a similar manner.

An advantage of this simplification is that it overcomes the problem of discontinuous heating rates. Each condenser refrigerant regime, vapour or two-phase, passesses a constant cooling rate. In addition, it allows the two-phase heat transfer analysis to be conducted with the entire length of tube. Suppose superheating were confined to a specific region, then tube temperatures bounding that region would have to be calculated. The effectiveness-NTU approach requires that the inlet and outlet temperature for each heat exchanger regime be known. Obviously, this is a more difficult approach than assuming consistent inlet and outlet flows for all regimes as does the simplication.

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* The term "condenser" is usely loosely. Generally, subcooling is accomplished in a separate heat exchanger. Here, condensing refers to the entire high temperature heat exchange operation. Note, also that the direction of refrigerant flow changes with either evaporator or condenser.

4.2.1. EVAPORATOR

The following equations describe the operation of an evaporator. Analysis is subdivided into a thermodynamic calculation for the single phase superheat vapour and a heat transfer approach for the more complicated two-phase operation.

Certain physical constraints directly influence the model. As a flooded evaporator, the evaporator tube bundles are considered totally submerged in liquid. The liquid being vaporized in the shell of the evaporator is liquid refrigerant. Water or another single phase fluid is considered to flow through the evaporator tubes.

i) Superheat Region

(5) $$Q_{SH} = M_R \, C_{PRv} \, \Delta T_{SH}$$

where Q_{SH} = superheat heat transfer
M_R = refrigerant mass flow
C_{PRv} = refrigerant vapour specific heat
ΔT_{SH} = superheat

$$\Delta T_{SH} = T_3 - ET$$

ii) Two-Phase Region

(6) $$Q_{EV} = (M_{SRC} - M_{SH}) \, C_{pw\ell} \, \varepsilon_E \, (T_1 - ET)$$

where Q_{EV} = two-phase heat transfer
M_{SRC} = coolant flow rate
M_{SH} = equivalent superheat coolant flow rate
$C_{pw\ell}$ = coolant specific heat
ε_E = evaporator effectiveness

(7)

$$M_{SH} = \frac{Q_{SH}}{C_{pw\ell}\,(T_1 - T_2)}$$

For a shell-and-tube heat exchanger with one shell pass and an even number of tube passes,

(8)

$$\varepsilon_E = 2\left\{\frac{1 + 1 + \exp\left[-NTU_E\right]}{1 - \exp\left[-NTU_E\right]}\right\}^{-1} \qquad [1]$$

where NTU_E = net transfer unit

(9)

$$NTU_E = \frac{U_E A_E}{(M_{SRC} - M_{SH})\, C_{PW\ell}}$$

where U_E = overall evaporator heat transfer coefficient

A_E = evaporator contact area

(10)

$$M_{SH} = \frac{Q_{SH}}{C_{PW\ell}\,(T_1 - T_2)}$$

$$A_E = N_E\, AEFF_E\, LEFF_E$$

where N_E = number of continuous evaporator tubes

$AEFF_E$ = effective external surface area of tube per unit length

$LEFF_E$ = effective length of tube

(11)

$$AEFF_E = \frac{AF_E + AR_E}{t_E + ROOT_E}$$

where AF_E = surface area of fin

AR_E = surface area of root

t_E = fin thickness

$ROOT_E$ = distance between adjacent fins

(12)

$$LEFF_E = N_{PEt}\, EDL$$

where N_{PEt} = number of tube passes

EDL = nominal evaporator length

(13)

$$U_E = \left\{ \frac{d_{EO}}{(d_{Ei})(h_{Ei})} + RFE + \frac{d_{EO}}{2\ k_m} \ln \frac{d_{EO}}{d_{Ei}} + \frac{1}{h_{EO}} \right\}^{-1}$$

where d_{EO} = tube outside diameter
d_{Ei} = tube inside diameter
R_{FE} = evaporator tube fouling factor
K_m = tube wall conductivity
h_{Ei} = tubeside convection coefficient
h_{EO} = shellside convection coefficient

If the flow in tube in laminar ($Re_{El} < 2300$)

(14)

$$H_{Ei} = \left(\frac{K_w}{d_{Ei}} \right) 3.66 \qquad [1]$$

where K_w = water conductivity

If the flow in tube is turbulent ($Re_{El} > 2300$)

(15)

$$h_{Ei} = \frac{K_w}{d_{Ei}}\ 0.023\ Re_{EI}^{\ 0.8}\ Pr^{\ 0.4} \qquad [1]$$

where Pr = PrantI number
RE_{EI} = internal tube reynolds number

(16)

$$Pr = \frac{C_{PWl}\ \mu_w}{K_w}$$

where μ_w = water viscosity

(17)

$$Re_{EI} = \frac{\left(\frac{M_{SRC}}{N_E} \right)}{\Pi\ D_{EI}\ \mu_w}$$

Assume an additive mechanism to determine shellside convection coefficient in the evaporator is valid,

(18)

$$h_{EO} = h_{EOB} + h_{EOC}$$

where h_{EOB} = boiling contribution to shellside convection coefficient

h_{EOC} = macro convection contribution to shellside convection coefficient

h_{EOB} is given by a transcendental equation of the form,

(19)

$$h_{EOB} = \left\{h_{EOV}^{4/3} + h_{EOR}\, h_{EOB}^{1/3}\right\}^{0.75} \qquad [1]$$

where h_{EOV} = micro convection contribution to boiling

h_{EOR} = radiation contribution to boiling

(20)

$$h_{EOV} = 0.62 \left[\frac{K_{RV}^{3}\,\rho_{RV}(\rho_{R\ell}-\rho_{Rv})g\,(H_{Rfg} + 0.4C_{PRv(TEPT-ET)})}{\mu_{Rv}\quad(TEP - ET)}\right]^{0.25} \qquad [1]$$

where K_{Rv} = refrigerant vapour conductivity

ρ_{Rv} = refrigerant vapour density

$\rho_{R\ell}$ = liquid refrigerant density

g = gravitational acceleration

h_{Rfg} = latent heat of refrigerant

TEPT = average tube surface temperature

μ_{Rv} = refrigerant vapour visocsity

(21)

$$h_{EOR} = \frac{\sigma\, \epsilon_{ET}\, (TEPT^4 - ET^4)}{(TEPT - ET)} \qquad [1]$$

where σ = Stefan - Boltzmann constant

ϵ_{ET} = evaporator tube emissivity

An initial guess of h_{EOB} is provided by,

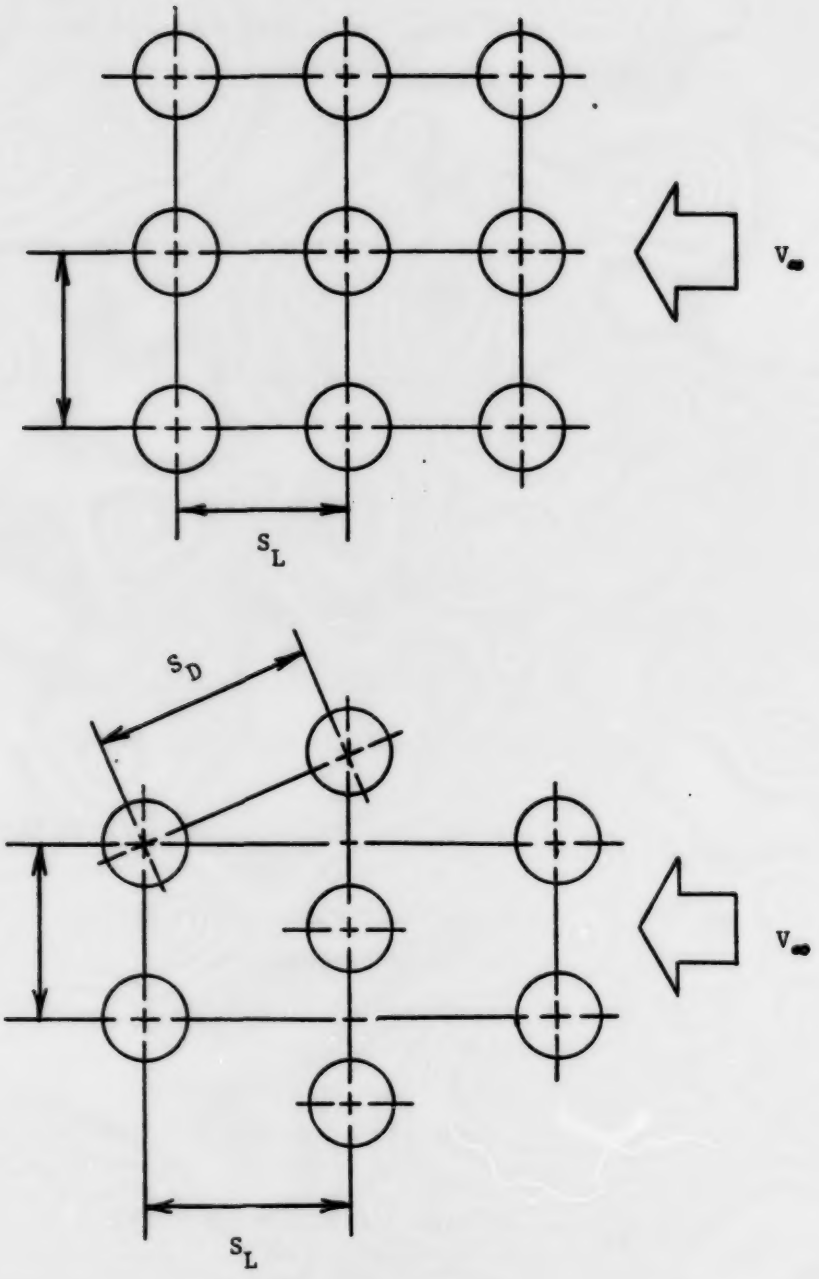

Fig (8): a. rectangular tube bundle arrangement b. triangular tube bundle arrangement

(22) $$h_{EOB} = h_{EOR} + h_{EOV}$$

(23)

$$h_{EOC} = \frac{K_{R\ell}}{D_{EO}}(C_{EO})\ Re_{EO}^{M_{EO}} \left(\frac{C_{PR\ell}\ \mu_{R\ell}}{K_{R\ell}}\right)^{0.36}\left(\frac{Pr_\infty}{Pr_s}\right)^{0.25} \qquad [1]$$

where $K_{R\ell}$ = refrogeramt liquid conductivity
C_{EO} = empirical constant
M_{EO} = empirical constant
Re_{EO} = external tube Reynolds number
$\mu_{R\ell}$ = liquid refrigerant viscosity
Pr_∞/Pr_s = ratio of freestream temperature and tube surface temperature Prantl numbers.

Assume that $\left(\frac{Pr_\infty}{P_{rs}}\right)^{0.25}$ equals unity and can be ignored.

(24) $$Re_{EO} = \frac{V_{Emax}\ d_{EO}\ \rho_{R\ell}}{\mu_{R\ell}} \qquad [1]$$

where V_{Emax} = maximum refrigerant velocity occurring within tube bundle

If tube bundle is arranged rectangularly,

(25) $$Ve_{max} = \frac{S_{TE}\ V_{\infty E}}{S_{TE} - d_{EO}} \qquad [1]$$

Where S_{TE} = transverse pitch , See. Fig. (8)
$V_{\infty E}$ = free stream velocity

If tube bundle is arrange triangularly,

(26) $$V_{Emax} = \frac{S_{TE}\ V_{\infty E}}{2(S_{DE}-d_{EO})} \qquad [1]$$

Where S_{DE} = diametral pitch; See Fig. (8)

(27)
$$V_{\infty E} = \frac{M_R \, v_{R\ell}}{\frac{\Pi}{4}\left(D_E^2 - NTOTAL_E \, d_{EO}^2\right)}$$

Where $NTOTAL_E$ = number of tubes through the cross-section

4.2.2 CONDENSER AND SUBCOOLER

HOT/WATER GENERATING CONDENSER

(i) Desuperheating Region

(28)
$$Q_{DSH} = M_R \, C_{PRv} \, (T_4 - CT)$$

Where Q_{DSH} = desuperheat heat transfer
M_R = refrigerant mass flow rate

C_{PRv} = refrigerant vapour specific heat

(ii) Subcooling Region

(29)
$$Q_{SC} = M_R \, C_{PR\ell} \Delta T_{SC}$$

Where Q_{SC} = subcool heat transfer
$C_{PR\ell}$ = refrigerant liquid specific heat

$\Delta T_{SC} = CT - T_5$

(iii) Two-Phase Region

(30)

$$Q_{CD} = (M_{SK} - M_{DSH} - M_{SC})\, C_{PW\ell}\, \varepsilon_c\, (CT - T_7)$$

Where Q_{CD} = two phase heat transfer
M_{SK} sink water flow rate
$C_{PW\ell}$ = water specific heat
ε_C = shell-and-tube heat exchanger effectiveness

For a shell-and-tube heat exchanger with one shell pass and and even number of tube passes,

(31)

$$\varepsilon_C = 2 \left\{1 + \frac{1 + \exp\left[- NTU_C\right]}{1 - \exp\left[- NTU_C\right]}\right\}^{-1} \qquad [1]$$

Where NTU_c = net transfer units of condenser

(32)

$$NTU_c = \frac{U_c\, A_c}{(M_{SK} - M_{SC} - M_{DSH})\, C_{PW\ell}}$$

(33)

$$M_{SC} = \frac{Q_{SC}}{C_{PW\ell}\, (T_8 - T_7)}$$

(34)

$$M_{DSH} = \frac{Q_{DSH}}{C_{PW\ell}\, (T_8 - T_7)}$$

(35)

$$A_c = N_c\, AEFF_c\, LEFF_c$$

Where N_c = number of continuous condenser tubes
$AEFF_c$ = effective external surface area of tube per unit length
$LEFF_c$ = effective length of tube

(36)

$$AEFF_c = \frac{AF_c + AR_c}{t_c + ROOT_C}$$

Where AF_c = surface area of fin
AR_c = surface area of root
t_c = fin thickness
$ROOT_C$ = distance between consecutive fins

(37)

$$LEFF_c = N_{pct}\ CDL$$

Where N_{pct} = number of tube passes
CDL = nominal condenser length

(38)

$$U_c = \left\{\frac{d_{co}}{d_{ci} \cdot h_{ci}} + RFC + \frac{d_{co}}{2\ k_m}\ \ln\ \frac{d_{co}}{d_{ci}} + \frac{1}{h_{co}}\right\}^{-1}$$

Where d_{co} = tube outside diameter
d_{ci} = tube indise diameter
RFC = condenser tube fouling factor
K_m = tube wall conductivity
h_{ci} = tubeside convection coefficient
h_{co} = shellside convection coefficient

If the flow in the tube is laminar ($Re_c < 2300$)

(39)

$$h_{ci} = \left(\frac{Kw}{d_{ci}}\right) 3.66 \qquad [1]$$

where Kw water conductivity

If the flow is turbulent ($Re_{ci} > 2300$)

(40)

$$h_{ci} = \frac{Kw}{d_{ci}} \left(0.023 \right) Re_{ci}^{0.8}\ Pr^{0.3}$$

Where Re_{ci} = Reynolds number
Pr = Prantl number

(41)

$$Pr = \frac{C_{PW\ell}\ \mu_w}{Kw}$$

Where μ_w = water viscosity

(42)

$$Re_{ci} = \frac{\left(\frac{M_{SK}}{N_c}\right)}{\Pi \, dci \, \mu_w}$$

(43)

$$h_{co} = 0.689 \left(\frac{K_{R\ell}^3 \, \rho_{R\ell}^2 \, g \, h_{RFG}}{\mu_{R\ell} \, DCEQ \, N_c}\right)^{\frac{1}{4}} \quad [2]$$

Where $K_{R\ell}$ = refrigerant liquid conductivity
$\rho_{R\ell}$ = refrigerant liquid density
g = gravitational acceleration
H_{RFG} = modified latent heat of refrigerant
$\mu_{R\ell}$ = refrigerant liquid viscosity
DCEQ = equivalent diameter

(44)

$$h_{RFG} = h_{Rfg} + C_v \, (T_4 - TCPT) \quad [2]$$

Where h_{Rfg} = latent heat of refrigerant
TCPT = average tube temperature

(45)

$$TCPT = \frac{T_7 + T_8}{2}$$

(46)

$$DCEQ = \frac{1.3 \, AF_c \, \eta}{AEFF_c \, (L_{mf})^{0.25}} + \frac{AF_c}{AEFF_c \, (d_{co})^{0.25}} \quad [2]$$

Where η = fin efficiency

(47)

$$\text{and Lmf} = \frac{\Pi}{4d_{cf}} \left(d_{cf}^{\,2} - d_{co}^{\,2}\right) \quad [2]$$

Where d_{cf} = Fin outside diameter [2]

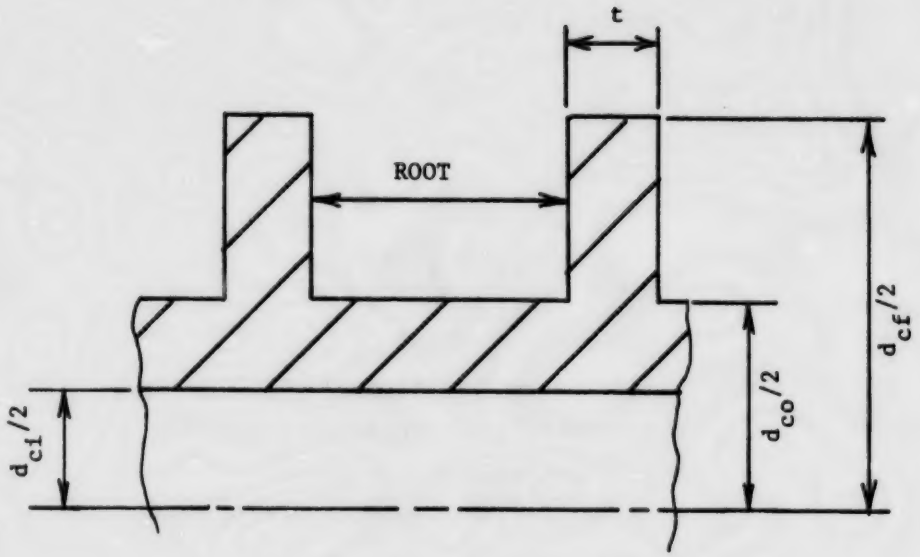

Fig (9): Fin detail

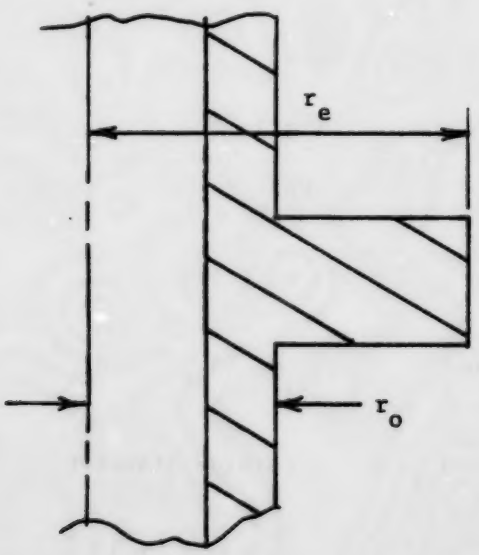

Fig (10): Fin detail

The fin efficiencies are calculated mathematically as follows from [3],

(48)

$$\eta = \frac{2\, r_o}{m(r_e^2 - r_o^2)} \left(\frac{I_1(mr_e)K_1(mr_o) - K_1(mr_e)I_1(mr_o)}{I_o(mr_o)K_1(mr_e) - I_1(mr_e)K_o(mr_o)} \right)$$

Where $I_n(\chi)$ are Modified Bessel Functions of order n

Accurate approximations $I_n(\chi)$ for $\chi < 10$

(49)

$$I_o(\chi) = \frac{1}{6}\left\{1 + \cosh(\chi) + 2\cosh\left(\frac{\sqrt{3}}{2}\chi\right) + 2\cosh\left(\frac{\chi}{2}\right)\right\}$$

(50)

$$I_1(\chi) = \frac{1}{6}\left\{\sinh(\chi) + \sqrt{3}\sinh\left(\frac{\sqrt{3}}{2}\chi\right) + \sinh\left(\frac{\chi}{2}\right)\right\}$$

(51)

Approximations to $K_o(\chi) = \int_o^\infty e^{-\chi \cosh t}\, dt$

(52)

$$\text{and } K_1(\chi) = \frac{-d}{d\chi}\left(K_o(\chi)\right)$$

(53)

$$K_o(\chi) = \frac{1}{4}\left\{e^{-\chi} + e^{-\chi\cosh(6)} + 2\sum_{J=1}^{\infty} \exp\left[-\chi\cosh(J/2)\right]\right\}$$

(54)

$$K_1(\chi) = \frac{1}{4}\left\{e^{-\chi} + \cosh(6)e^{-\chi\cosh(6)} + 2\sum_{J=1}^{\infty} \cosh(J/2)\exp\left[-\cosh(J/2)\right]\right\}$$

(55) For annular fins, $m = \left(\frac{h_{co}}{K_m(t_c/2)}\right)^{0.5}$

Where h_{co} = initial estimate of h_{co}

By the superpositin of effects,

(56)

$$h_{co} = h_{cod} + h_{cov}$$

Where h_{cod} = convection coefficient for film condensation on horixontal tube banks

h_{cov} = convection coefficient for a bank of horizontal tubes in a fluid flow

(57)

$$h_{cod} = 0.728 \quad \frac{g\,\rho_{R\ell}\,(\rho_{R\ell}-\rho_{Rv})\,K_{R\ell}^{3}\,H_{fg}}{\mu_{R\ell}(CT-TCPT)\,DCO} \qquad [1]$$

(58)

$$h_{cov} = \frac{K_{Rv}}{d_{co}} \quad C_o(Re)_{Dmax}^{M_o} \quad Pr^{0.36}\left(\frac{Pr_\infty}{Pr_s}\right)^{\frac{1}{4}} \qquad [1]$$

Where Pr, Pr_∞ = free stream Prantt number

Pr_s = Prantt number evaluated at pipe surface temp.

C_o = constant dependent upon Reomax

M_o = constant dependent up Reomax

(59)

Assume that $\left(\frac{Pr_\infty}{Pr_s}\right)^{0.25} \simeq 1.0$

$$Re_{Dmax} = \frac{\rho_{Rv} V_{maxc} d_{co}}{\mu_{Rv}}$$

Where μ_{Rv} = refrigerant vapour viscosity

V_{maxc} = maximum fluid velocity occuring within the tube bank

If tubes are aligned rectangularly

(60)

$$Vmaxc = \frac{V_{\infty c} S_{tc}}{S_{tc} - d_{co}} \qquad [1]$$

Where S_{tc} = transverse pitch

$V_{\infty c}$ = free stream velocity

If tubes are arrranged triangularly,

(61)

$$Vmaxc = \frac{V_{\infty c} S_{tc}}{2(S_{DC} - d_{co})} \qquad [1]$$

Where S_{DC} = diametral pitch

(62)

$$V_{\infty c} = \frac{M_R V_{R\ell}}{\pi(D_c^2 - NTOTAL_c d_{co}^2)}$$

Where $NTOTAL_c$ = number of tubes through the cross-section.

STEAM GENERATING CONDENSER

The analysis of the steam generating condenser is similar to the hot water condenser of the previous section. In fact, superheat and subcool heat transfer rates are calculated in the same manner. Moreover, in keeping with the heat exchanger simplification previously presented, only part of the total water* flow is considered to be effected by the two phase regime in the shell.

The cause for a differing analysis results from the water's discontinuous heating rate (MC_p) as it begings a two-phase regime. By analysing each water flow separately (liquid, two-phase mixture and vapour) the diffulty is overcome.

Before proceeding, the existence of sufficient heat transfer to allow the water coolant to attain saturation must be verified. Compare the heat transfer supposedly occurring assuming only liquid flowing through the tubes, Q_{SAT}', with the actual thermodynamic requirement, Q_{SAT}'',

(63)
$$Q_{SAT}' = (M_{SK} - M_{SC} - M_{DSH})\, C_{pw\ell}\, E_c\, (CT-T_7)$$

(64)
$$Q_{SAT}'' = (M_{SK} - M_{SC} - M_{DSH})\, C_{pw\ell}\, (T_p-T_7)$$

Where
- M_{SK} = sink coolant flow rate
- M_{SC} = equivalent coolant flow rate causing refrigerant to subcool
- M_{DSH} = equivalent coolant flow rate causing refrigerant vapour to de-superheat
- $C_{PW\ell}$ = liquid specific heat
- T_p = coolant liquid saturation temperature
- T_7 = coolant inlet temperature

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* In discussing the steam generating condenser, "water" becomes a blanket term referring to both liquid and vapour phases simultaneously.

Note that tube side properties should be based only on the inlet temperature, T_7, rather than an average of T_7 and T_8. Since T_7 is supposedly close to saturation, an upper limit on T_8, the mathematical convergence scheme may produce an estimate for T_8 for which properties do not exist.

If $Q_{SAT'}$ is less than $Q_{SAT"}$, then insufficient heat is transferred from the refrigerant to the coolant and the analysis is subsequently that of the hot water generating condenser. Having established that liquid saturation is achieved, determine the necessary condenser contact area required. This is done using a simple Newton - Raphson procedure,

(65)

$$A_c^{n+1} = A_c^{n} - \frac{f(AC^n)}{f'(AC^n)}$$

Where A_c^{n} = the current value of contact area
A_c^{n+1} = next estimate of contact area

(66)

$$F(A_c^{n}) = Q_{SAT'} - Q_{SAT"}$$

(67)

$$F'(A_c^{n}) = \frac{f(A_c^{n} - \alpha A_c^{n}) - f(A_c^{n})}{- A_c}$$

Where $0 < \alpha < 1$ is a multiplication factor

In the above cases, the contact area effects Q'_{sat} through the heat exchanger's "net transfer units" and subsequently effectiveness.

Once the contact area has been established, the associated length of tubing is found,

(68)

$$LEFF_c' = \frac{A_c}{N_c \; AEFF_c}$$

Where $LEFF_c'$ = length of tubing required to reach saturation

A_c = required contact area

$AEFF_c$ = effective contact area per unit length

Using this information, calculate pressure drop occurring in the liquid portion of the condenser tubes,

(69)

$$\Delta P_{TC} = \frac{M_{SK}^2 \, V_{w\ell}}{2 \, N_c^2 \, (\frac{\Pi}{4} d_{ci}^2)^2} \left(\frac{f_{TC} \, LEFF'_C}{d_{ci}} + 0.4 \right)$$

Where f_{TC} = friction factor

ΔP_{TC} = pressure loss in tubing

Note that the "0.4" accounts for the contraction of the flow at the inlet.

(70)

$$f_{TC} = \left\{ - 2.0 \log \left(\frac{\varepsilon_{ct}}{3.7 \, d_{ci}} + \frac{2.51}{Re_{ci} \left(f_{TC} \right)^{0.5}} \right) \right\}^{-2.0} \qquad [5]$$

Where ε_{ct} = condenser tube surface roughness

Re = Reynolds number for the liquid in condenser tube

An initial estimate for f_{TC} is furnished by,

(71)

$$f_{TC} = 1.02 \, (\log Re_{ci})^{-2.5} \qquad [5]$$

(72)

$$Re_{ci} = \frac{4 \, M_{SK}}{N_c \, \Pi \, d_{ci} \, \mu_{w\ell}}$$

Having determined ΔP_{TC}, a new saturation point is defined as shown in figure (11).

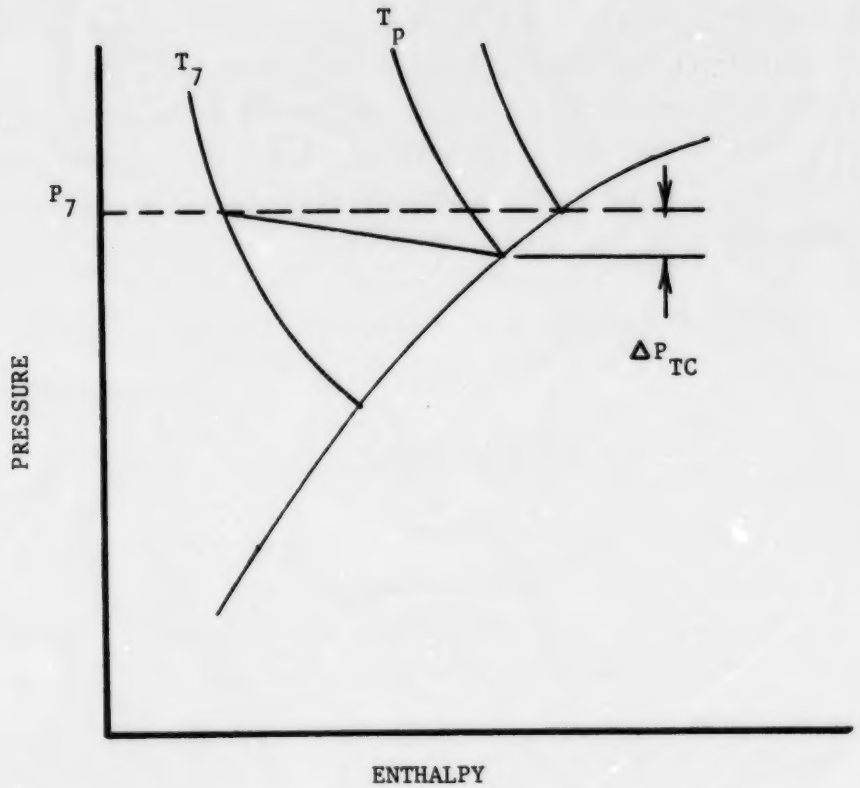


Fig (11): The effect of single-phase condenser tubeside pressure drop on the determination of saturation temperature T_p.

Assume that upon reaching saturation, the tube temperature will remain constant until the coolant's vapour saturation point is reached. Both fluids on either side of the tube are experiencing phase transformation and possess infinitely large heating rates. Consequently, the effectiveness - NTU approach wrongly predicts a heat exchanger effectiveness of zero and is unusable.

To determine the resulting heat transfer, the following procedure is proposed,

(73)

$$Q_{tp}' = \int_a^b U(x)\ (CT-WCT)dA = I$$

Where $U(x)$ = local overall heat transfer coefficient
dA = increment in contact area
$a,b,$ = limits of two phase tubing length
WCT = water condensing temperature

In the above situation, $U(x)$ varies in accordance with the local quality of the mixture which is a function of position, that is,

(74)

$$U(x) = \left\{ \frac{d_{co}}{d_{ci}HCI'} + R_{fc} + \frac{d_{co}}{2\ Km} \ln\left(\frac{d_{co}}{d_{ci}}\right) + \frac{1}{h_{co}} \right\}^{-1}$$

Where are variables have been previously defined except, HCI', the two phase internal convection coefficient)

(75)

$$HCI' = K1\ (CO)^{K2}(25\ Fr_{\ell})^{K5}h_{\ell} + K3\ (BO)^{K4}(25\ Fr_{\ell})^{K6}h_{\ell}F_{f1} \qquad [4]$$

Where K1 to K6 constants dependent upon Fr_{ℓ} and Co See Table (2)
$F_{f\ell}$ = 1.0 is a fluid dependent multiplication factor,
h_{ℓ} = single phase liquid heat transfer coefficient
BO = boiling number
CO = convection number
Fr_{ℓ} = froude number

(76)

$$h_\ell = 0.023 \left(\frac{G(1-\chi)D}{\mu_\ell}\right)^{0.8} Pr^{0.4} \frac{K_{w\ell}}{d_{ci}} \qquad [4]$$

Where G = mass flux

(77)

$$BO = \frac{g''}{GH_{wfg}} \qquad [4]$$

Where g" = heat flux

(78)

$$CO = \left(\frac{\chi}{1-\chi}\right)^{-0.8} \left(\frac{P_{wv}}{P_{w\ell}}\right)^{0.5} \qquad [4]$$

To obtain these equations, the local quality must be known. Hence, HCI' is obtained through an iterative procedure by using values of the previous or adjacent point and improving thereafter.

Starting from the position where liquid saturation is first attained, assume that at an incremental distance downstream, Δx, the quality is still zero. Calculate the associated heat flux for that point. Assuming heat flux varies linearly with distance, the incremental heat transfer for that increment of tube area is given by,

(79)

$$Q_{\Delta x} = \pi d_{ci} \left(\left(\frac{U_o - U_i}{2} \right) \Delta x + U_i \Delta x \right)$$

Where U_o = local heat transfer coefficient at outlet

U_i = local heat transfer coefficient at inlet

Obtain a new estimate for quality as,

(80)

$$\chi(x) = \frac{Q_{\Delta x} + (Q_{\Delta x})_{sum}}{H_{wfg}}$$

Where $(Q_{\Delta x})$ sum = cumulative sum of heat transfer

H_{wfg} = latent heat of water

Recalculate the local heat transfer coefficient using new quality and repeat until convergence is established. Similarly, march down the length of tube until either the end of pipe is reached, or the quality becomes zero. By saving the value of the local heat transfer coefficient, the value of I may be calculated using Simpson's rule,

(81)

$$I = \frac{L}{3n}\left(U_a + U_b + 4 \sum_{\substack{i=1 \\ odd}}^{n-1} U_j + 2 \sum_{\substack{i=2 \\ even}}^{n-1} U_j \right) \qquad [9]$$

Where n = an equal even number of subintervals into which the interval from a to b is divided

L = two-phase length of tubing

(82)

$$L = b - a$$

Begin at n=10 and repeat for n=20, if tolerance, say 0.1%, is not achieved repeat doubling n each time until reached.

If the end of the pipe has not been reached at the conclusion of the two-phase procedure, superheating of water vapour will occur. This regime of heat exchange is similar to the subcool stage. Assuming vapour flows at this point,

(83)

$$A''_{CDS} = (M_{SK} - M_{SC} - M_{DSH})\, C_{PWv}\, \varepsilon_c\, (CT - T_q)$$

Where ε_c is obtained as for the hot water condenser except that vapour properties are used instead.

T_q = vapour saturation point temperature

To determine T_q, the two-phase pressure drop must be calculated. Since the losses associated with tube returns are minor [2], consider only straight tube losses. Pierre [10] correlated results for R-12 and R-22. It is assumed that these results are applicable to water flow as well:

(84)

$$\Delta P_{TC}'' = \left(\frac{f_{TC}'' \; L}{d_{ci}} + \frac{\Delta\chi}{\chi_m} \right) G^2 \; . v_m \qquad [10]$$

Where f_{TC}'' = two-phase friction factor
$\Delta\chi$ = quality change
χ_m = mean quality
v_m = mean specific volume
L = two-phase tube length

(85)

$$v_m = V_{w\ell} + \chi_m \, (v_{wv} - v_{vi}) \qquad [10]$$

Where v_{wv} = specific volume
$v_{w\ell}$ = liquid specific volume

(86)

$$\chi_m = 0.5 \, (\chi_a + \chi_b) \qquad [7]$$

(87)

$$\Delta\chi = \chi_b - \chi_a$$

Where χ_a = quality at start of two-phase region
χ_b = quality at end of two-phase region

(88)

$$f_{TC}'' = 0.0185 \left(\frac{K_f}{Re_f} \right)^{0.25} \qquad [10]$$

(89)

$$K_f = \frac{J \; H_{wfg} \; \Delta\chi}{L \quad g} \qquad [7]$$

(90)

$$Re_f = \frac{4 \; M_{SK}}{\Pi \; d_{ci} \; \mu_{w\ell} \; Nc} \qquad [10]$$

Diagrammatically, the new vapour saturation point temperature is obtained as shown in Figure (11).

4.3 EXPANSION DEVICE

Expansion devices can be modelled as simple as non-linear equations relating pressure drop with liquid refrigerant bolume flux. Appendix (D) contains an example of a typical expansion value.

The equation for the expansion device is of the form,

(90)

$$\log (\Delta Px) = A_o + A_1 \log (GPMx)$$

Where A_o and A_1 are coefficients dependent upon the unit

ΔPx = pressure drop across expansion device

GPMx = Volume flux (liquid)

4.4 PRESSURE LOSSES MODULE

This routine accounts for the pressure losses which occur through the refrigerant and coolant cycles.

4.4.1 HEAT EXCHANGER SHELLSIDE LOSSES

Shellside pressure losses are negligible through both the condenser and evaporator. This is a result of circulation and baffle considerations. Baffles tend to increase pressure drop by forcing the fluid to continuously cross the tube bundles. However, baffles are not used in evaporators or condensers using halogen refrigerants and thus the pressure drop is small. Even in baffled evaporators, according to Kern [6], slow circulation in a flooded evaporator is responsible for negligible pressure loss.

4.4.2 PIPING SYSTEM

In this model, each major component in the refrigerant cycle is linked through a simple piping scheme as shown in figure (4). *

Losses are the result of pipe friction and momentum losses associated with pipe directional changes. This model assumes that a specified number of 90° bends exist in each section. Although this may not actually describe the real arrangement, it accounts for minor losses through an equivalent configuration. The total head loss is given by,

(92)

$$\Delta h_{total} = h_f + \Sigma h_m$$

---

* In some systems, the expansion valve discharges directly into the evaporator. Although figure (4) displays pipe section 6 as having a finite length, it can be simulated by setting the physical length of pipe 6 to zero.

Where Δh_{total} = total refrigerant head loss

h_f = frictional head loss

Σh_m = sum of all minor losses from pipe bends

(93)

$$\Sigma h_m = (NB_n)\, K_n$$

Where n = 3,4,5 or 6 in accordance with section in question

NB_n = specified number of elbows in that section

K_n = K-value or resistance coefficient for 90° elbow

A correlation for resistance coefficients given in [5] is presented below,

(94)

$$K_n = 0.4680342 - 0.2189008 \log (D_n)$$

Where n = 3,4,5 or 6 in accordance with section in question

D_n = pipe diameter in inches

(95) Frictional head loss is given by,

$$h_f = f \frac{L_n V^2}{2 g D_n} \qquad [5]$$

Where n = 3,4,5, or 6 in accordance with section in question

h_f = frictional head loss

f = Darcy - Wersbach friction factor

L_n = straight-line, length of section

g = gravitational acceleration

V = average pipe velocity

(96)

$$V = \frac{M_R v_R}{\frac{\Pi}{4} D_n^2}$$

Where v_R = refrigerant specific volume

If the flow is laminar (Re < 2300),

(97)

$$f = \frac{64}{Re} \qquad [5]$$

If the flow is turbulent (Re > 2300), as a first approximation use,

(98)

$$f = 1.02 \ (\log Re)^{-2.5} \qquad [5]$$

Then iterate the following equation until convergence occurs,

(99)

$$f = \left\{ -2.0 \log \left(\frac{\varepsilon / D_n}{3.7} + \frac{2.51}{(Re) f^{0.5}} \right) \right\}^{-2} \qquad [5]$$

Where ε = pipe roughness

The Reynolds number is given by,

(100)

$$Re = \frac{4 \ M_R}{\pi \ D_n \ \mu_R}$$

Where μ_R = refrigerent viscosity

The previous pipe relations require that lengths, diameters, and materials already be known before calculation of losses can proceed. Normally, in selecting heat pump components, this information is not specified. Accordingly, the following discussion proposes a method for sizing the piping system.

A few arbitrary assumptions are first made. To fix surface roughness (ε), assume all pipe used is drawn and nearly smooth (ε = .000005"). Based on existing heat pumps, supply a default pipe length of 5' for each section of pipe. These values can always be changed since they are input parameters.*

Most importantly, due to the iterative nature of the method presented, an initial pressure gradient estimate is required. In agreement with common industrial practice, estimates have been developed for this purpose based on [12].

(101)

Suction Line : $\left(\frac{\Delta P_3}{L_3}\right)_{nom} = 0.002 + \frac{0.028(ET-399)}{210}$, (psia)

(102)

Discharge Line : $\left(\frac{\Delta P_4}{L_4}\right)_{nom} = 0.02 + \frac{0.03(CT-534)}{60}$, (psia)

(103) Subcooler

Discharge: : $\left(\frac{\Delta P_5}{L_5}\right)_{nom} = 0.02$, (psia)

Where $\frac{\Delta Pn}{L_n}$ = nominal pressure gradient for a given pipe section

The nominal head loss for a pipe section is,

(104)

$$h_{fn} = \left(\frac{\Delta P_n}{L_n}\right)_{nom} \frac{L_n}{\rho_R g} + 0.26\ NB_n$$

Where ρ_R = refrigerant density

g = gravitational acceleration

NB_n = arbitrary number of 90° elbows

Unfortunately, the nominal pressure gradient is insenstive to variation in refrigerant conditions and hence annot be accepted as a constant.

---

* refers to list of default parameters whose values can be changed on the discretion of the system user.

The required pipe diameter is given by,

(105)

$$D_n = \frac{4\ M_R}{\Pi\ Re_n\ \mu_R}$$

Where Re_n = Reynolds number for unknown pipe diameter

Since the optimum operating conditions are not known, use the furnished initial estimates for ET and CT as references for property data.

A reasonable initial estimate of Reynold's number is provided by,

(106)

$$Re'_n = 1.43\ \beta^{0.416} \qquad [5]$$

(107)

$$\beta = \left\{ \frac{128\ g}{\Pi\ L_n} \frac{h_{fn}\ M_R^3}{\mu_R^5\ v_R^2} \right\}^{0.5} \qquad [5]$$

Where h_{fn} = nominal head loss

From the above equation, it is clear that the nominal head loss is a weak factor of the Reynolds number and use of it is justifiable.

If $Re'_n < 2300$, the flow is laminar and,

(108)

$$Re'_n = \left(\frac{\beta}{8}\right)^{0.5} \qquad [5]$$

If $Re'_n > 2300$, the flow is turbulent and,

(109)

$$Re'_n = \left\{ -2.0\ \beta \log \left(\frac{\Pi\varepsilon\ \mu_R\ (Re'_n)}{M_R\ (14.8)} + \frac{2.51}{\beta}\ (Re'_n) \right)^{1.5} \right\}^{0.4}$$

The above relations dealing with the design of the pipe should only be conducted once at the beginning of each simulation execution in the absence of better data. At this point, the piping configuration by which other refrigerant conditions will be subject is set.

4.4.3. HEAT EXCHANGER TUBESIDE LOSSES

The calculations presented in this section are pertinent to both single-phase coolant flows. The loss involved with steam generation is closely linked with the heat transfer calculation and appears in section 4.2.2

The head loss is given as,

(110)
$$h_f = \frac{V^2}{2g}\left(\frac{f\,L}{d} + 0.8 + K_n\,(N_{pt} - 1)\right)$$

Where
- V = average fluid velocity
- f = Darcy - Weisbach friction factor
- g = gravitational acceleration
- L = straight-line tube length
- d = internal tube diameter
- K_n = pipe bend resistance factors
- N_{pt} = number of tube passes

(111)
$$V = \frac{4\,m\,v}{\Pi\,d^2}$$

Where
- m = mass flow through tube
- v = specific volume

The friction factor, f, is given in equation (99). the initial estimate for f is provided in equation (98). The Reynolds number application is also presented with equation (100). K_n is defined using equation (94).

Summarizing, these calculations determine the pressure drop through the evaporator tubes and in the single phase portion of the condenser coolant if applicable. These are ΔP_{TE} and ΔP_{TSC}, respectively.

4.4.4 APPLICATION

By submitting estimate values of ET, CT, ΔT_{SH} and ΔT_{SC}, a portion of a simplified refrigerant cycle is in fact established (see Fig. (12)). After execution of the compressor model, the discharge temperature is provided and the refrigerant cycle is complete except for pressure losses which are calculated separately. See figures (13,14,15,16).

This section diagramitically indicates how the various pressure drops calculated in this section modify the statepoints.

The individual pressure drops which can be calculated are,

ΔP_5 = subcooled liquid line pressure loss
ΔP_3 = suction line pressure loss
ΔP_4 = discharge line pressure loss
ΔP_6 = expansion valve outlet line pressure loss

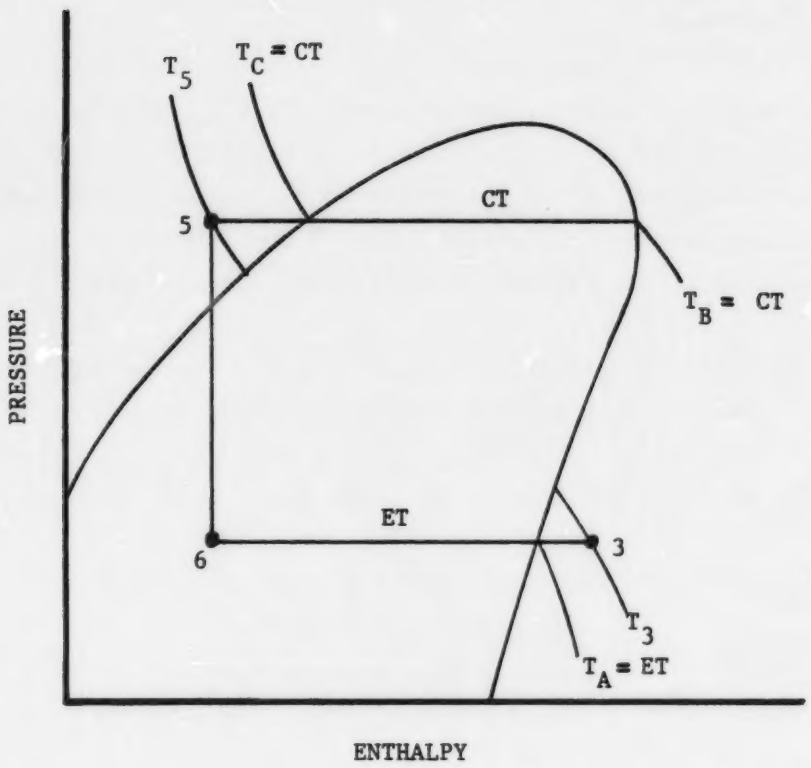
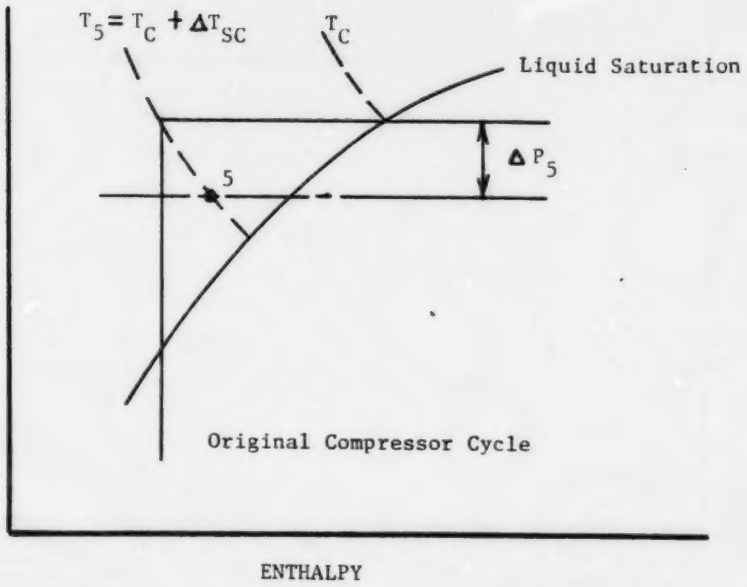


Fig (12): Approximate estimate of the refrigerant cycle resulting from the compressor model



Fig(13): Statepoint 5 alteration resulting from pressure drop in condenser discharge line

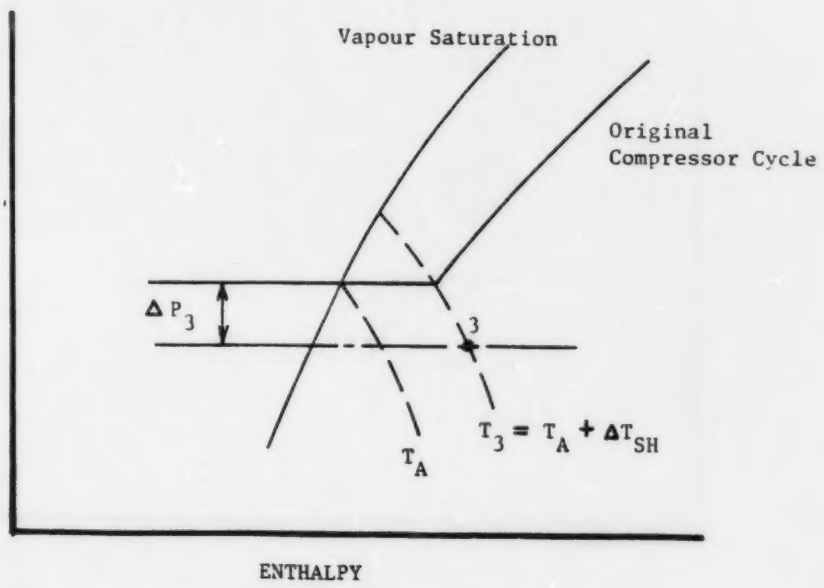
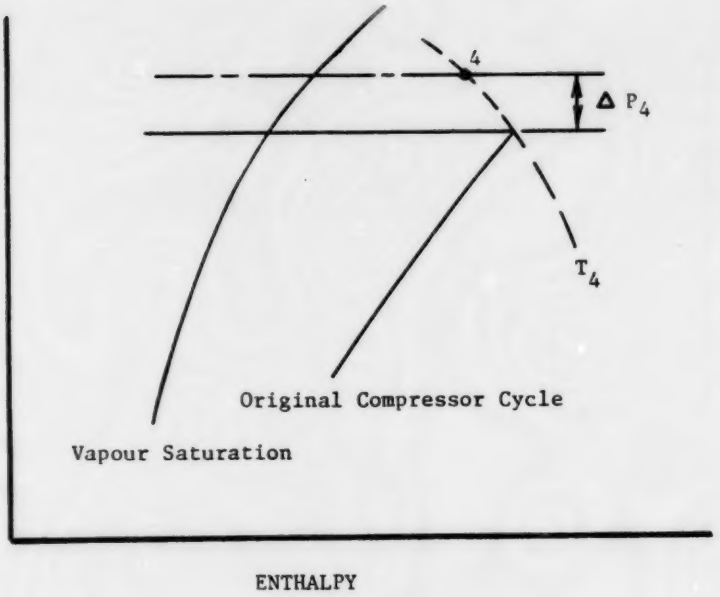
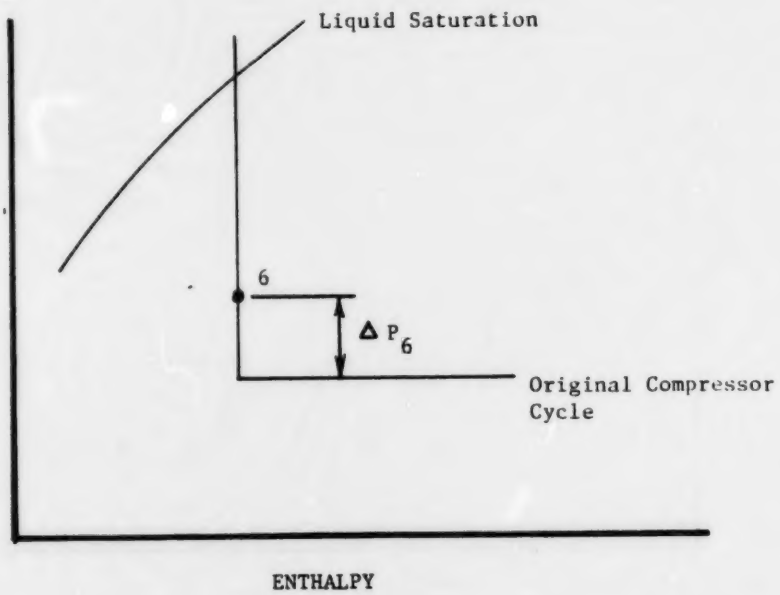


Fig (14): Statepoint 3 alteration resulting from pressure drop in suction line



Fig(15): Statepoint 4 alteration resulting from pressure drop in discharge line



Fig(16): Statepoint 6 alteration resulting from pressure drop in expansion device discharge line

5.0 SYSTEM ARCHITECTURE

This section presents the same material as section four. Whereas the previous section omits some degree of detail in favour of clairty, this section is complete. It is designed to aid the systems analyst or computer programmer in producing working code. To this effect, several features are evident: specific nomenclature, consistent variable labels, definite property referencing, unit consistency, special function identification and logic flow charting.

5.1 NOMENCLATURE

This section is presented as an aid to understanding the many equations appearing in section 5.0. It explicitly describes each variable. The variables themselves are classified as input parameters or simple variables. Input parameters are those requiring values prior to execution. Typically, they describe the physical characteristics such as dimensions. the simple variables are internal variables and bear no consequence to the user.

Each variable is presented with its proper units under which the equations and conversion factors have been formulated.

INPUT PARAMETERS

EDL - nominal evaporator length (ft)

D_E - evaporator internal shell diameter (in)

d_{Ei} - evaporator tube inside diameter (in)

d_{Eo} - evaporator tube outside diameter (in)

d_{EF} - evaporator tube outside fin diameter (in)

$ROOT_E$ - distance inbetween consecutive fins on evaporator tube (in)

t_E - evaporator tube fin thickness (in)

S_{DE} - evaporator tube bundle diametral pitch (in)

S_{TE} - evaporator tube bundle transverse pitch (in)

S_{LE} - evaporator tube bundle longitudinal pitch (in)

N_{PET} - number of tube passes in evaporator

F_{TE} - flag; if "on", denotes triangular evaporator remove bundle arrangement

N_E - number of returning tubes in evaporator

$NTOTAL_E$ - number of tubes in an evaporator cross-section (i.e. $NTOTAL_E = (N_{PET})(N_E)$)

R_{fE} - evaporator tubeside fouling factor (hft^2R/Btu)

CDL - nominal condenser length (ft)

D_c - condenser internal shell diameter (in)

d_{ci} - condenser tube inside diameter (in)

d_{co} - condenser tube outside diameter (in)

d_{CF} - condenser tube outside fin diameter

$ROOT_c$ - distance inbetween consecutive fins on condenser tubes (in)

t_c - condenser tube fin thickness (in)

S_{DC} - condenser tube bundle diametral pitch (in)

S_{TC} - condenser tube bundle transverse pitch (in)

S_{LC} - condenser tube bundle longitudinal pitch (in)

N_{PCT} - number of tube passes in condenser

F_{TC} - flag; if "on", denotes triangular condenser bundle arrangement

N_c - number of returning tubes in evaporator

$NTOTAL_c$ - number of tubes in a condenser cross-section (i.e. $NTOTAL_c = (N_{PCT})(N_c)$

F_{fc} - condenser tubeside fouling factor (hft^2R/Btu)

NB_3 to NB_6 - number of 90° elbows per respective pipe section

L_3 to L_6 - length of each respective pipe section

ε_3 to ε_6 - surface roughness of each respective pipe section (ft)

R - refrigerant gas constant (ft.lbf/lbm/R)

r_e - condenser tube outside fin radius (in)
r_o - condenser tube outside radius (in)

ε_{ET} - evaporator tube emissivity

ε_{CT} - condenser tube emissivity

VARIABLES

Note that any variable with a single numeric subscript from 1 to 8 can be referenced to specific statepoint or location in the heat pump cycle. See fig. (4). The variables with numberic subscripts are,

T - temperature (R)
P - pressure (Psia)
h - enthalpy (Btullbin)
v - specific volume (ft^3/lbm)
D - pipe diameter (in)
h_f - nominal pipe headloss (ft of refrigerant)
β - dimensionless parameter used in determing pipe Reynolds number
R_e' - pipe Reynods number estimate
R_e - actual pipe Reynolds number
f - pipe friction factor
K - pipe resistance factor
ΔP - pressure loss is given pipe sectio (psia)
h_6' - enthalpy of refrigerant liquid saturation point at CT (Btu/lbm)

h_6' - enthalpy of refrigerant liquid saturation point at ET (Btu/lbm)

x_6' - quality at statepoint 6

Other variables with alphanumeric subscripts are now defined,

TCPT - average tubeside temperature in condenser (R)

TCPSI - average shellside vapour temperature in condenser (R)

TCPS2 - average shellside liquid temperature in subcooler (R)

TEPT - average tubeside temperature in evaporator (R)

TEPF - evaporator film property temperature (R)

ΔT_{SC} - subcool (R)

ΔT_{SH} - superheat (R)

ET - nominal evaporation temperature (R)

CT - nominal condensation temperature (R)

T_{5S} - refrigerant liquid saturation temperature at the pressure of statepoint 5 (R)

T_p - condenser coolant liquid saturation temperature (R)

T_q - condenser coolant vapour saturation temperature (R)

h_{Ei} - evaporator tubeside convection coefficient (Btu/h/ft^2/R)

h_{EOR} - evaporator radiation contribution to boiling heat transfer (Btu/h/ft^2/R)

h_{EOV} - evaporator shellside micro convection contribution to boiling (Btu/h/ft^2/R)

h_{EOB} - boiling contribution to shellside convection coefficient (Btu/h/ft^2/R)

h_{EOC} - macro convection contribution to shellside convection coefficient (Btu/h/ft^2/R)

h_{EO} - evaporator shellside convection coefficient (Btu/h/ft^2/R)

h_{ci} - condenser tubeside convection coefficient (Btu/h/ft^2/R)

h_{COD} - convection coefficient for film condensation on

horizontal tube banks (Btu/h/ft^2/R)

h_{COV} - convection coefficient for a bank of horizontal tubes in a fluid flow (Btu/h/ft^2/R)

Δh_c - modified form of latent heat (Btu/h/ft^2/R)

h_{co}' - initial estimate for h_{co} (Btu/h/ft^2/R)

h_{co} - condenser shellside convection coefficient (Btu/h/ft^2/R)

h_ℓ - two-phase convection coefficient for tubeside flow (Btu/h/ft^2/R)

h_p - coolant enthalpy at liquid saturation (Btu/h/ft^2/R)

h_q - coolant enthalpy at vapour saturation (Btu/h/ft^2/R)

h_{ci}' - condenser tubeside convection coefficient for two-phase flow (Btu/h/ft^2/R)

$LEFF_E$ - effective evaporator tube length (ft)

AF_E - surface area of a fin (in^2)

AR_E - surface area of root (in^2)

$AEFF_E$ - effective evaporator surface area per unit length (in)

A_E - total evaporator surface area (ft^2)

$LEFF_C$ - effective condenser tube length (ft)

AF_C - surface area of a condenser fin (in^2)

AR_C - surface area of a condenser root (in^2)

$AEFF_c$ - effective condenser surface area per unit length (in)

d_{CEQ} - equivalent condenser tube diameter (in)

A_C - total condenser surface area (ft^2)

ΔA_{CDS} - incremental condenser tube steam generating area (ft^2)

$LEFF_C'$ - effective condenser tube length required for single phase coolant flow (ft^2)

L_{REM} - remaining condenser tube length next $LEFF_c'$ (ft^2)

Re_{EI} - evaporator tubeside Re

Re_{CI} - condenser tubeside Re

Re_{EO} - evaporator shellside Re

Re_{CO} - condenser shellside Re

Re_{CI}' - condenser tubeside liquid saturation Rℓ

f_{TS} - evaporator tubing friction factor

f_{TC} - condenser tubing friction factor
f_{TC}' - condenser tubing two-phase friction factor

$\varepsilon_c, \varepsilon_c^*$ - condenser effectiveness
ε_E - evaporator effectiveness
X - coolant quality in condenser tubes
X_m - mean condenser coolant quality
X_x - condenser coolant quality at end of two-phase flow
X_o - condenser coolant quality at beginning of two-phase flow
M - fin efficiency parameter
VAR1 - fin efficiency parameter
VAR2 - fin efficiency parameter
I_{11} - fin efficiency parameter
I_{12} - fin efficiency parameter
I_{02} - fin efficiency parameter
K_{02} - fin efficiency parameter
K_{11} - fin efficiency parameter
K_{12} - fin efficiency parameter

α - fraction used to obtain ΔA_{CPS}
CO - convection number
BO - boiling number
Fr_ℓ - liquid froude number in condenser coolant
G - mass flux (lbm/hr/ft^2)
M_R - refrigerant flow rate riven by compressor model (lbm/hr)
M_x - refrigerant flow rate given by expansion value model (lbm/hr)
M_{SK} - total sink coolant flow rate (lbm/hr)
M_{SRC} - total source coolant flow rate (lbm/hr)
M_{SC} - equivalent coolant flow rate through subcooler (lbm/hr)
M_{SH} - equivalent coolant flow rate causing superheating (lbm/hr)

M_{DSH} - equivalent coolant flow rate causing de-superheating (lbm/hr)

J = 778.16 (ft.lbf/btu)

g = 32.17 (ft/sec^2)

ΔP_{TE} - evaporator coolant pressure loss (psia)

ΔP_{TSC} - condenser single-phase coolant pressure loss (psia)

ΔP_{TC} - single-phase coolant pressure loss in a steam-generating condenser (psia)

$\Delta P_{TC}''$ - condenser coolant two-phase pressure drop (psia)

P - brake horsepower (Btu/h)

OHR - Oil heat rejection (Btu/h)

Q_{in} - evaporator coolant thermodynamic heat transfer (Btu/h)

Q_{out} - condenser coolant thermodynamic heat transfer (Btu/h)

Q_C - condenser fundamental heat transfer (Btu/h)

Q_E - evaporator fundamental heat transfer (Btu/h)

Q_{CD} - condenser two phase heat transfer (Btu/h)

I - condenser steam generating heat transfer (Btu/h)

Q_{CDS}'' - condenser steam superheat heat transfer (Btu/h)

Q_{SC} - subcool heat transfer (Btu/h)

Q_{DSH} - desuperheat heat transfer (Btu/h)

Q_{SH} - superheat heat transfer (Btu/h)

Q_{EV} - evaporator two-phase heat transfer (Btu/h)

q'' - condenser heat flux ($Btu/h/ft^2$)

K_f - Pierre's two-phase factor

ΔX - tube length increment (ft)

$V_{\infty E}$ - evaporator shellside free steam Velocity (ft/hr)

$V_{\infty C}$ - condenser shellside free steam velocity (ft/hr)

V_{maxE} - max velocity in evaporator shell (ft/hr)

V_{maxC} - max velocity in condenser shell (ft/hr)

C_o - coefficienty; table (1)

M_o - coefficient; table (1)

V_E - evaporator overall heat transfer coefficient ($Btu/h/ft^2/R$)

U_c - condenser overall heat transfer coefficient ($Btu/h/ft^2/R$)

U_{cx} - local condenser overall heat transfer coefficient

(Btu/h/ft^2/R)

NTU_E - evaporator net transfer units

NTU_c, NTU_c* - condenser net transfer units

5.2. SPECIAL FUNCTIONS

The relations presented in the succeeding sections make use of special functions. These separately called routines are used to provide refrigerant and coolant property data to the various models. Output for these routines can be obtained either from tabular lookup and interpolation routines or correlations. This section provides a summary of these functions and identifies those for whom correlations exist.

The following notation is used:

T = temperature input

P = pressure input

v = specific volume input

→ = means that a correlation is available; see Appendix (A)

REFRIGERANT

→ $C_{PRv}[T]$ - * specific heat of saturated vapour (Btu/lbm•R)

where R is the gas constant

→ $C_{PR\ell}[T]$ - specific heat of saturated liquid (btu/lbm•R)

→ $v_{Rv}[T,P]$ - specific volume of vapour (ft^3/lbm)

→ $v_{R\ell}[T]$ - specific volume of saturated liquid (ft^3/lbm)

$\mu_{Rv}[T]$ - saturated vapour visosity (lbm/hr•ft)

$\mu_{R\ell}[T]$ - saturated liquid viscosity (lbm/hr•ft)

---

* This value is, in fact, obtained as : $C_{PRv} = C_{vRv} + R$;

+ $C_{PRv}[T,v]$ - specific heat at constant volume of vapour (Btu/lbm•R)
+ $h_{Rfg}[T]$ - latent heat of vaporization (Btu/lbm)
+ $h_{Rv}[T,c]$ - enthalpy of vapour (Btu/lbm)
+ $P_{RSAT}[T]$ - saturation pressure given temperature (psia)
$T_{RSAT}[P]$ - saturation temperature given pressure (°R)
$K_{Rv}[T]$ - conductivity of saturated vapour (Btu/hr•ft•R)
$K_{R\ell}[T]$ - conductivity of saturated liquid (Btu/hr•ft•R)

COOLANT

$C_{PWv}[T]$ - specific heat of saturated vapour (Btu/lbm•R)
$C_{PW\ell}[T]$ - specific heat of saturated liquid (Btu/lbm•R)
$v_{wv}[T]$ - specific volume of saturated vapour (ft^3/lbm)
$v_{w\ell}[T]$ - specific volume of saturated liquid (ft^3/lbm)
$\mu_{wv}[T]$ - saturated vapour viscosity (lbm/hr•ft)
$\mu_{w\ell}[T]$ - saturated liquid viscosity (lbm/hr•ft)
$h_{wfg}[T]$ - latent heat vaporization (Btu/lbm)
$P_{WSAT}[T]$ - saturation pressure given temperature (psia)
$T_{WSAT}[T]$ - saturation temperature given pressure (°R)
$K_{wv}[T]$ - conductivity of saturated vapour (Btu/hr•ft•R)
$K_{w\ell}[T]$ - conductivity of saturated liquid (Btu/hr•ft•R)
$Pr[T]$ - Prantt number of saturated liquid (-)

OTHER

$K_{mc}[T]$ - conductivity of condenser tube metal (Btu/hr•ft•R)
$K_{ME}[T]$ - conductivity of evaporator tube metal (Btu/hr•ft•R)

5.3. RESIDUAL EQUATIONS

The functions which will be submitted to the chosen non-linear simultaneous equation solver are called "residentual" equations. These equations all take the form,

$$F_n(\chi_1, \chi_2, \ldots \chi i) = o$$

The eight equations are,

$$F_I = 0 = Q_{in} + P - (Q_{out} + OHR)$$
$$F_{II} = 0 = Q_{in} - M_{SRC}\,(C_{pw\ell}[TEPT]\,(T_1 - T_2) - v_{R\ell}[TEPT]\,\Delta P_{TSE})$$

If the condenser is hot water generating,

$$F_{III} = 0 = Q_{out} - M_{SK}(C_{pw\ell}[TCPT](T_8 - T_7) - v_{R\ell}[TCPT]\Delta P_{TSC})$$

$$F_{III} = 0 = Q_{out} - M_{SK}(C_{pw\ell}[.5(T_7+T_8)](T_p - T_7) + (h_q - h_p) + C_{pwv}[T_q](T_8 - T_q)]$$
$$F_{IV} = 0 = M_R - M_x$$
$$F_V = 0 = M_R\,(h_4 - h_5) - Q_c$$
$$F_{VI} = 0 = M_R\,(h_3 - H_6) - Q_E$$
$$F_{VII} = 0 = Q_c - Q_{out}$$
$$F_{VIII} = 0 = Q_E - Q_{in}$$

5.4. OTHER EQUATIONS

The equations listed in this seation are those which must be solved in order to obtain values for the variables used in the residual equations. For simplicity they are presented in a format different from the residual equations.

The order of presentation will vary from that of section four. The current sequesce, to be used if being coded, insures that no unknowns are encountered progressing down the list. Section 5.2.4 will contain flow charts utilizing these equations.

Note that certain conventions are used:

* is a multiplication marker used to delimit conversion factors or incomplete lines of multiplication

( ) normal term delimiters

[ ] function brackets; delimit the argument as belonging to function immediately preceeding.

PIPE DESIGN

(P.1) $$h_{f3} = \left(\frac{\Delta P_3}{L_3}\right)_{nom} \frac{L_3 V_{RV}[ET+0.5\Delta T_{SH}]}{g} + 0.26$$

(P.2) $$h_{f4} = \left(\frac{\Delta P_4}{L_4}\right)_{nom} \frac{L_4 V_{RV}[CT+10]}{g} + 0.26$$

(P.3) $$h_{f5} = \left(\frac{\Delta P_5}{L_5}\right)_{nom} \frac{L_5 V_{R\ell}[CT-0.5\Delta T_{SC}]}{g} + 0.26$$

(P.4) $$\beta_3 = \left(\frac{128\, g\, h_{f_3} M_R^3}{\pi L_3 \mu_{Rv}[ET+0.5\Delta T_{SH}] V_{Rv}[ET+0.5\Delta T_{SH}]} * 3600^2\right)^{0.5}$$

(P.5) $$\beta_4 = \left(\frac{128\, g\, h_{f_4} M_R^3}{\pi L_4 \mu_{Rv}[CT+10] V_{RV}[CT+10]} * 3600^2\right)^{0.5}$$

(P.6) $$\beta_5 = \left(\frac{128\, g\, h_{f_5} M_R^3}{\pi L_5 \mu_{R\ell}[CT-0.5\Delta T_{SC}] V_{R\ell}[CT-0.5\Delta T_{SC}]} * 3600^2\right)^{0.5}$$

(P. 7) $$Re_3 = 1.43\, \beta_3^{0.416}$$

(P. 8) $$Re_4 = 1.43\, \beta_4^{0.416}$$

(P. 9) $$Re_5 = 1.43\, \beta_5^{0.416}$$

(P.10) $Re_3 = \left(\frac{\beta_3}{8}\right)^{0.5}$

(P.11) $Re_4 = \left(\frac{\beta_4}{8}\right)^{0.5}$

(P.12) $Re_5 = \left(\frac{\beta_5}{8}\right)^{0.5}$

(P.13) $$Re_3 = \left\{-2.0\beta_3 \log\left(\frac{\Pi\varepsilon_3\mu_{RV}[ET+0.5\Delta T_{SH}]Re_3}{14.8\ M_R} + \frac{2.51(Re_3)^{1.5}}{\beta_3}\right)\right\}^{0.4}$$

(P.14) $$Re_4 = \left\{-2.0\beta_4 \log\left(\frac{\Pi\varepsilon_4\mu_{RV}[CT+10]\ Re_4}{14.8\ M_R} + \frac{2.51(Re_4)^{1.5}}{\beta_4}\right)\right\}^{0.4}$$

(P.15) $$Re_5 = \left\{-2.0\beta_5 \log\left(\frac{\Pi\varepsilon_5\mu_{R\ell}[CT-0.5\Delta T_{SC}]Re_5}{14.8\ M_R} + \frac{2.51(Re_5)^{1.5}}{\beta_5}\right)\right\}^{0.4}$$

(P.16) $$D_3 = \frac{4\ M_R}{\Pi\ Re_3\ \mu_{Rv}[ET+0.5\Delta T_{SH}]} * 12$$

(P.17) $$D_4 = \frac{4\ M_R}{\Pi\ Re_4\ \mu_{Rv}[CT+10]} * 12$$

(P.18) $$D_5 = \frac{4\ M_R}{\Pi\ Re_5\ \mu_{R\ell}[CT-0.5\Delta T_{SC}]} * 12$$

(P.19) $D_6 = D_5$

PRESSURE DROPS - EVAPORATOR COOLANT

(PD.1) $TEDT = 0.5\ (T_1+T_2)$

(PD.2) $$Re_{EI} = \frac{4\ M_{SRC}}{N_E\Pi\ d_{EI}\mu_{w\ell}[TEPT]} * 12$$

(PD.3) $f_{TE} = 1.02\ (\log Re_{EI})^{-2.5}$

$$(PD.4) \qquad f_{TE} = \left(-2.0 \log\left(\frac{\varepsilon_{ET}}{3.7 d_{EI}} * 12 + \frac{2.51}{Re_{EI} f_{TE}^{0.5}}\right)\right)^{-2}$$

$$(PD.5) \qquad \Delta P_{TE} = \frac{M_{SRC}^2 V_{w\ell}[TEPT]}{2 N_E^2 \frac{\Pi}{4} d_{EI}^2 z} \left(\frac{f_{TE} LEFF_E}{d_{EI}} * 12 + 0.8\right) * \frac{1728}{3600^2}$$

PRESSURE DROPS - CONDENSER COOLANT (SINGLE PHASE)

$$(PD.6) \qquad TCPT = T_7$$

$$(PD.7) \qquad TCPT = 0.5 \left(T_7 + T_{RSAT}[P_7]\right)$$

$$(PD.8) \qquad Re_{CI} = \frac{4 M_{SK}}{N_C \Pi d_{CI} \mu_{w\ell}[TCPT]} * 12$$

$$(PD.9) \qquad f_{TC} = 1.02 \left(\log Re_{CI}\right)^{-2.5}$$

$$(PD.10) \qquad f_{TC} = \left(-2.0 \log\left(\frac{\varepsilon_{CT}}{3.7 d_{CI}} * 12 + \frac{2.51}{Re_{EI} f_{TE}^{0.5}}\right)\right)^{-2}$$

$$(PD.11) \qquad \Delta P_{TC} = \frac{M_{SK}^2 V_{w\ell}[TCPT]}{2 N_c^2 \frac{\Pi}{4} d_{ci}^2 z} \left(\frac{f_{TC} LEFF_C}{d_{CI}} * 12 + 0.8\right) * \frac{1728}{3600^2}$$

PRESSURE DROPS - PIPING SYSTEM

$$(PD.12) \qquad Re_3 = \frac{4 M_R}{\Pi D_3 \mu_{Rv}[ET + 0.5 \Delta T_{SH}]}$$

$$(PD.13) \qquad Re_4 = \frac{4 M_R}{\Pi D_4 \mu_{Rv}[0.5 (CT + T_4)]}$$

$$(PD.14) \qquad Re_5 = \frac{4 M_R}{\Pi D_5 \mu_{R\ell}[CT - 0.5 \Delta T_{SC}]}$$

$$(PD.15) \qquad f_3 = 1.02 \left(\log Re_3\right)^{-2.5}$$

(PD.16) $f_4 = 1.02\ (\log Re_4)^{-2.5}$

(PD.17) $f_5 = 1.02\ (\log Re_5)^{-2.5}$

(PD.18) $f_3 = \left\{-2.0 \log \left(\frac{\varepsilon_3}{3.7D_3} * 12 + \frac{2.51}{Re_3 f_3^{0.5}}\right)\right\}^{-2}$

(PD.19) $f_4 = \left\{-2.0 \log \left(\frac{\varepsilon_4}{3.7D_3} * 12 + \frac{2.51}{Re_4 f_4^{0.5}}\right)\right\}^{-2}$

(PD.20) $f_5 = \left\{-2.0 \log \left(\frac{\varepsilon_5}{3.7D_5} * 12 + \frac{2.51}{Re_5 f_5^{0.5}}\right)\right\}^{-2}$

(PD.21) $k_3 = 0.4680342 - 0.2189008 \log (D_3)$

(PD.22) $k_4 = 0.4680342 - 0.2189008 \log (D_4)$

(PD.23) $k_5 = 0.4680342 - 0.2189008 \log (D_5)$

(PD.24) $\Delta P_3 = \frac{M_R^2\ \mu_{Rv}[ET+0.5\Delta T_{SH}]}{2\left(\frac{\Pi}{4} D_3^2\right)^2}\left(\frac{f_3 L_3}{D_3} * 12 + K_3(NB_3)\right) * \frac{1728}{3600^2}$

(PD.25) $\Delta P_4 = \frac{M_R^2\ \mu_{Rv}[0.5(CT+T_4)]}{2\left(\frac{\Pi}{4} D_4^2\right)^2}\left(\frac{f_4 L_4}{D_4} * 12 + K_4(NB_4)\right) * \frac{1728}{3600^2}$

(PD.26) $\Delta P_5 = \frac{M_R^2\ \mu_{R\ell}[CT-0.5\Delta T_{SC}]}{2\left(\frac{\Pi}{4} D_5^2\right)^2}\left(\frac{f_5 L_5}{D_5} * 12 + K_5(NB_5)\right) * \frac{1728}{3600^2}$

(PD.27) $h_6' = h_{Rv}[CT, v_{Rv}[CT, P_{RSAT}[CT]]] - h_{Rfg}[CT] - C_{PR\ell}[CT-0.5\Delta T_{SC}]\Delta T_{SC}$

(PD.28) $h_6'' = h_{Rv}[ET, v_{Rv}[ET, P_{RSAT}[ET]]] - h_{Rfg}[ET]$

(PD.29) $X_6' = 1 - \left(\frac{h_6' - h_6''}{h_{Rfg}[ET]}\right)$

(PD.30) $$Re_6 = \frac{(1-\chi_6')M_R}{(\frac{\Pi}{4} D_6)\mu_{R\ell}[ET]}$$

(PD.31) $$f_6 = 1.02\ (\log Re_6)^{-2.5}$$

(PD.32) $$f_6 = \{-2.0 \log (\frac{\epsilon_6}{3.7 D_6} * 12 + \frac{2.51}{Re_6 f_6^{0.5}})\}^{-2}$$

(PD.33) $$K_6 = 0.4680342 - 0.2189008 \log (D_6)$$

(PD.34) $$\Delta P_6 = \frac{M_R^2(v_{Rv}[ET]\chi_6' + v_{R\ell}[ET](1-\chi_6'))}{2\ (\frac{\Pi}{4} D_6^2)^2}(\frac{f_6 L_6}{D_6} * 12 + K_6(NB_6)) * \frac{1728}{3600^2}$$

STATEPOINT ALTERATIONS - REFRIGERANT CYCLE

(SA.1) $$T_5 = CT - \Delta T_{SC}$$

(SA.3) $$P_5 = P_{RSAT}[CT] - \Delta P_5$$

(SA.4) $$T_{5s} = T_{RSAT}[P_5]$$

(SA.5) $$h_5 = h_{5s}[T_5s,\ v_{R\ell}[T_{5s}]] - h_{Rfg}[T_{5s}] - C_{PR\ell}[0.5(T_{5s}+T_5)] * (T_{5s} - T_5)$$

(SA.7) $$T_3 = ET + \Delta T_{SH}$$

(SA.8) $$P_3 = P_{RSAT}[ET] - \Delta P_3$$

(SA.9) $$v_3 = v_{Rv}[T_3,\ P_3]$$

(SA.10) $h_3 = h_{Rv}[v_3, T_3]$

(SA.11) $P_4 = P_{RSAT}[CT]+\Delta P_4$

(SA.12) $v_4 = v_{Rv}[T_4, P_4]$

(SA.13) $h_4 = h_{Rv}[T_4, v_4]$

(SA.14) $h_6 = h_5$

EVAPORATOR

(E.1) $Q_{SH} = M_R C_{PRv}[0.5(T_3+ET)]\Delta T_{SH}$

(E.2) $TEPF = 0.5(TEPT+ET)$

(E.3) $h_{Ei} = \frac{K_{w\ell}[TEPT]}{d_{Ei}}\ 3.66 * 12$

(E.4) $h_{Ei} = \frac{k_{w\ell}[TEPT]}{d_{Ei}}\ (0.023)Re_{ei}^{0.8}\ Pr[TEPT]^{0.4} * 12$

(E.5) $h_{EOR} = \frac{0.1714(10^{-8})^{\varepsilon}ET(TEPT^4-ET^4)}{(TEPT-ET)}$

(E.6) $H_{EOV} = \{1.5552(10^8) * \frac{K_{Rv}[TEPF]^3}{v_{R\ell}[TEPF]}(\frac{1}{v_{R\ell}[ET]} - \frac{1}{v_{Rv}(TEPF]})$

$* g\frac{(h_{Rfg}[ET]+0.4C_{PRv}[TEPF](TEPF-ET)}{\mu_{Rv}[TEPF]d_{EO}(TEPT-ET)}\}^{0.25}$

(E.7) $H_{EOB} = H_{EOR} + h_{EOV}$

(E.8) $h_{EOB}^{n+1} = (h_{EOV}^{4/3} + h_{EOR}\ (h_{EOB}^{n})^{1/3})\ 0.75$

$$(E.9) \qquad V_{\infty E} = \frac{4\, M_R\, v_{R\ell}\,[ET]}{\pi(D_E^2 - NTOTAL_E\, d_{EO}^2)} * 144$$

$$(E.10) \qquad V_{EMAX} = \frac{S_{TE}\, V_{\infty E}}{S_{TE} - d_{EO}}$$

$$(E.11) \qquad V_{EMAX} = \frac{S_{TE}\, V_{\infty E}}{2(S_{DE} - d_{EO})}$$

$$(E.12) \qquad Re_{EO} = \frac{V_{EMAX}\, d_{EO}}{v_{R\ell}[ET]\,\mu_{R\ell}[ET]} * \frac{1}{12}$$

$$(E.13) \qquad h_{EOC} = \frac{K_{R\ell}[ET]}{d_{EO}}\, C_o\, Re_{EO}^{M_o} \left\{\frac{C_{PR\ell}[ET]\,\mu_{R\ell}[ET]}{K_{R\ell}[ET]}\right\}^{0.36} *12$$

TABLE (1)

	Re	C_o	M_o	[1]
Aligned	10^3-2 x 10^5	0.27	0.63	
Staggered (S_I/S_L<2)	10^3-2 x 10^5	$0.35(S_T/S_L)^{1.5}$	0.60	
Staggered (S_T/S_L>2)	10^3-2 x 10^5	0.40	0.60	
Aligned	2 x 10^5-2 x 10^6	0.021	0.84	
Staggered	2 x 10^5-2 x 10^6	0.022	0.84	

(E.14) $h_{EO} = h_{EOB} + h_{EOC}$

(E.15) $V_E = \left\{\frac{1}{h_{EO}} + \frac{1}{h_{E1}}\left(\frac{d_{EO}}{d_{E1}}\right) + \frac{1}{12} * \frac{D_{EO}}{K_{ME}} \ln\left(\frac{d_{EO}}{d_{E1}}\right) + R_{fE}\right\}^{-1}$

(E.16) $LEFF_E = N_{PET}\ EDL$

(E.17) $AF_E = \frac{\Pi}{4}\left((d_{EF} + t_E)^2 - d_{EO}{}^2\right)$

(E.18) $AR_E = \Pi\ d_{EO}\ ROOT_E$

(E.19) $AEFF_E = \frac{AF_E + AR_E}{t_E + ROOT_E}$

(E.20) $A_E = N_E\ AEFF_E\ LEFF_E * \frac{1}{12}$

(E.21) $M_{SH} = \frac{Q_{SH}}{C_{pw\ell}[TCPT](T_1 - T_2)}$

(E.22) $NTU_E = \frac{U_E\ A_E}{(M_{SRC} - M_{SH})C_{pw\ell}[TEPT]}$

(E.23) $\epsilon_E = 2\left\{1 + \frac{1 + \exp(-NTU_E)}{1 - \exp(-NTU_E)}\right\}^{-1}$

(E.24) $Q_{EV} = (M_{SRC} - M_{SH})\ C_{pw\ell}\ [TCPT]\ \epsilon_E\ (T_1 - ET)$

(E.25) $Q_E = Q_{EV} + Q_{SH}$

CONDENSER - HOT/WATER GENERATING

(C.1) $TCPS1 = 0.5\ (CT + T_4)$

(C.2) $TCPS2 = 0.5\ (CT + T_5)$

(C.3) $Q_{DSH} = M_R\ C_{PRv}\ [TCPS1]\ (T_4 - CT)$

(C.4) $Q_{SC} = M_R\ C_{PR\ell}\ [TCPS2]\ (CT - T_5)$

(C.5) $h_{ci} = \frac{K_{w\ell}\ [TCPT]}{d_{ci}}\ (0.023)\ Re_{ci}^{0.8}\ P_r[TCPT]^{0.4} * 12$

(C.6) $h_{ci} = \frac{k_{w\ell}[TCPT]}{d_{ci}}\ (3.66) * 12$

(C.7) $h_{COD} = 0.728\ \{(1.552(10^8) * \frac{1}{v_{R\ell}[CT]}\ (\frac{1}{v_{R\ell}[CT]} - \frac{1}{v_{Rv}[TCPS1]}) * K_{R\ell}[CT]^3 h_{Rfg}[CT])/(N_c \mu_{R\ell}[CT](CT-TCPT)d_{co})\}^{0.25}$

(C.8) $V_{\infty c} = \frac{4\ M_R\ v_{R\ell}[CT]}{\Pi(D_c^{\ 2} - NTOTAL_c\ d_{co}^{\ 2})} * 144$

(C.9) $V_{cmax} = \frac{S_{TC}\ V_{\infty C}}{S_{TC} - D_{CD}}$

(C.10) $V_{cmax} = \frac{S_{CT}\ V_{\infty C}}{2(S_{DC} - d_c)}$

(C.11) $Re_{co} = \frac{V_{cmax}\ d_{co}}{\mu_{Rv}[TCPS1]\ V_{Rv}[TCPS1]} * \frac{1}{12}$

(C.12) $$h_{cov} = \frac{K_{Rv}[TCPS1]C_o}{d_{co}} Re_{co}^{M_o} \left\{\frac{C_{prv}[TCPS1]\mu_{Rv}[TSCS1]}{K_{Rv}[TCPS1]}\right\}^{0.36} * 12$$

See table (1) for coefficients C_o, M_o

(C.13) $$h_{co}' = h_{COD} + h_{COV}$$

(C.14) $$M = \left\{\frac{2\, h_{co}'}{K_{Rv}[TCPS1]\, t_c}\right\}^{0.5}$$

(C.15) $$VAR1 = M_{re}/12$$

(C.16) $$VAR2 = M_{ro}/12$$

(C.17) $$I_{11} = \frac{1}{6}\{\sinh(VAR1) + \sqrt{3}\sinh[\sqrt{3}\,VAR1/2) + \sinh(VAR1/2)\}$$

(C.18) $$I_{12} = \frac{1}{6}\{\sinh(VAR2) + \sqrt{3}\sinh(\sqrt{3}\,VAR2/2) + \sinh(VAR2/2)\}$$

(C.19) $$I_{02} = \frac{1}{6}\{1+\cosh(VAR2) +2\cosh(\sqrt{3}\,VAR2/2)+ 2\cosh(VAR2/2)\}$$

(C.20) $$K_{02} = \frac{1}{4}\{\exp(-VAR2)+ \exp(-VAR2\cosh 6)+2\sum_{J=1}^{n}\exp(-VAR2\cosh(\tfrac{J}{2})\}$$

(C.21) $$K_{11} = \frac{1}{4}\{\exp(-VAR1)+ \cosh 6\, \exp(-VAR1\cosh 6) + 2\sum_{j=1}^{n}\cosh(J/2)\exp(-VAR1\cosh(J/2))\}$$

(C.22) $$K_{12} = \frac{1}{4}\{\exp(-VAR2) + \cosh 6\exp(-VAR2\cosh 6) + 2\sum_{J=1}^{n}\cosh(J/2)\exp(-VAR2\cosh(J/2))\}$$

(C.23) $$\eta = \frac{d_{co}}{M(r_e^2 - r_o^2)} \frac{(I_{11}K_{12} - K_{11}I_{12})}{(I_{02}K_{11} - I_{11}K_{02})} * \frac{1}{12}$$

(C.24) $$AR_c = \Pi\, d_{co}\, ROOT_c$$

(C.25) $$AF_c = \Pi \left((r_e + t_c/2)^2 - r_o^2\right)$$

(C.26) $$AT_c = AF_c / \left(t_c + ROOT_c\right)$$

(C.27) $$AEFF_c = \left(AF_c + AR_c\right) / \left(t_c + ROOT_c\right)$$

(C.28) $$d_{CEQ} = \left\{ \frac{1.3\ \eta\ AF_c}{AEFF_c \left(\frac{\Pi}{d_{CF}} \left(d_{CF}^2 - d_{CO}^2\right)\right)^{0.25}} + \frac{AF_c}{AEFF_c\ d_{co}^{0.25}} \right\}^{-4}$$

(C.29) $$\Delta h_c = h_{Rfg}[CT] + C_{vRv}[CT]\ (T_4 - CT)$$

(C.30) $$h_{co} = 0.689 \left\{ \frac{1.552\ (10^8) * K_{R\ell}[CT]^3\ g\ \Delta h_c}{d_{ceq}\ v_{R\ell}[CT]^2 \mu_{R\ell}[CT] N_C^{0.5}} \right\}^{0.25}$$

(C.31) $$U_c = \left\{ \left(\frac{AT_c \ln(d_{co}/d_{ci})}{2\ \Pi\ K_m\ [TCPT]} + \frac{AT_c}{\Pi\ d_{ci} h_{ci}} + \frac{AT_c\ R_{fc}}{\Pi\ d_{ci}} \right) * \frac{1}{2} + \frac{AT_c}{AEFF_c\ h_{co}} \right\}^{-1}$$

(C.32) $$LEFF_c = N_{PCT}\ CDL$$

(C.33) $$A_c = N_c\ AEFF_c\ LEFF_c\ * \frac{1}{12}$$

(C.34) $$M_{SC} = \frac{Q_{SH}}{C_{pw\ell}\left[T_8 - T_7\right]}$$

(C.35) $$M_{DSH} = \frac{Q_{DSH}}{C_{pw\ell}\left[T_8 - T_7\right]}$$

(C.36) $$NTU_C = \frac{U_c\ A_c}{\left(M_{SK} - M_{SC} - M_{DSH}\right) C_{pw\ell}\left[TCPT\right]}$$

(C.37) $$\varepsilon_c = 2 \left\{ 1 + \frac{1 + \exp\left[-NTU_c\right]}{1 - \exp\left[-NTU_c\right]} \right\}^{-1}$$

$$(C.38) \qquad Q_{CD} = (M_{SK} - M_{SH} - M_{SDH})\, C_{pw\ell}[TCPT]\, \varepsilon_c\, (CT - T_7)$$

$$(C.39) \qquad Q_c = Q_{CD} + Q_{SC} + Q_{DSH}$$

CONDENSER - STEAM GENERATING

The steam generating condenser analysis requires that equations (C.1) to (C.40) be executed. The only alteration to make is:

$$TCPT = T_7$$

The explicit steam generating set of equations follows:

$$(CS.1) \qquad Q_{CDS}' = (M_{SK} - M_{SC} - M_{DSH})C_{pw\ell}[TCPT]\varepsilon_c(CT-T_7)$$

$$(CS.2) \qquad Q_{CDS}'' = (M_{SK} - M_{SC} - M_{DSH})C_{pw\ell}[TCPT](T_{WSAT}[P_7]-T_7)$$

$$(CS.3) \qquad \Delta A_{CDS} = \alpha\, A_c$$

$$(CS.4) \qquad A_c^{n+1} = A_c^{n} \left(\frac{Q_{CDS}''}{Q_{CDS}'}\right)$$

$$(CS.5) \qquad NTU_c = \frac{U_c\, A_c^{n+1}}{(M_{SK} - M_{SC} - M_{DSH})C_{pw\ell}[TCPT]}$$

$$(CS.6) \qquad NTU_c^* = \frac{V_c\, (A_c^{n+1} + \Delta A_{CDS})}{(M_{SK} - M_{SC} - M_{DSH})C_{pw\ell}[TCPT]}$$

$$(CS.7) \qquad \varepsilon_c = 2 \left\{ 1 + \frac{1 + \exp(-NTU_c)}{1 - \exp(-NTU_c)} \right\}^{-1}$$

$$(CS.8) \qquad \varepsilon_c^* = 2 \left\{ 1 + \frac{1 + \exp(-NTU_c^*)}{1 - \exp(-NTU_c^*)} \right\}^{-1}$$

(CS.9) $$A_c^{n+1} = A_c^{n} - \frac{\Delta A_{CDS}\ (\varepsilon_c(CT-T_7)-(T_p - T_7))}{(\varepsilon_c^{*}(CT-T_7)-(Tp - T_7))}$$

(CS.10) $$LEFF'_c = \frac{A_c^{n+1}}{N_c AEFF_c} * 12$$

(CS.11) $$Re_{ci} = \frac{4\ M_{SK}}{N_c\ \Pi\ d_{ci}\ \mu_{w\ell}[TCPT]} * 12$$

(CS.12) $$f_{Tc} = 1.02\ (\log Re_{ci})^{-2.5}$$

(CS.13) $$f_{Tc} = \left\{-2.0\ \log\left(\frac{\varepsilon_{CT}}{3.7\ d_{ci}} * 12 + \frac{2.51}{Re_{ci} f_{TC}^{0.5}}\right)\right\}^{-2.0}$$

(CS.14) $$\Delta P_{TC} = \frac{M_{SRC}^{2}\ v_{w\ell}[TCPT]}{2N_c^{2}\left(\frac{\Pi}{4} d_{ci}^{2}\right)^{2}}\left(\frac{f_{TC}\ LEFF'_c}{d_{ci}} * 12 + 0.4\right) * \frac{1728}{3600^{2}}$$

(CS.15) $$T_p = T_{WSAT}\ [P_7 - \Delta P_{TC}]$$

(CS.16) $$L_{REM} = LEFF_c - LEFF'_c$$

(CS.17) $$\Delta X = \frac{L_{REM}}{10}$$

(CS.18) $$CO = \left(\frac{X^{n}}{1-X^{n}}\right)^{-0.8}\left(\frac{v_{R\ell}\ [T_p\ ]}{v_{Rv}\ [T_p\ ]}\right)^{0.5}$$

(CS.19) $$q'' = U_c^{n}\ (CT - T_p)$$

(CS.20) $$G = \frac{4\ (M_{SK} - M_{SC} - M_{DSH})}{\Pi\ d_{ci}^{2}} * 144$$

(CS.21) $$BO = \frac{q''}{G\ H_{Rfg}\ [T_p]}$$

$$(CS.22) \qquad h_{\ell} = 0.023\left(\frac{G(1-\chi^{n})d_{ci}}{\mu_{w\ell}[T_p]}*\frac{1}{12}\right)^{0.8} Pr_{w\ell}[T_p]^{0.4}\ \frac{K_{w\ell}[T_p]}{d}*12$$

$$(CS.23) \qquad Fr_{\ell} = \frac{G^2\ v_{Rw\ell}[T_p]^2}{g\ d_{ci}} * \frac{12}{3600^2}$$

$$(CS.24) \qquad h'_{ci} = K1\ (CO)^{K2}(25Fr_{\ell})^{K5} + K3(BO)^{K4}(25Fr_{\ell})^{K6}h_{\ell}$$

TABLE (2)

	$Fr_{\ell} < 0.04$		$Fr_{\ell} > 0.04$	
CONSTANT	CO < 0.65	CO > 0.65	CO < 0.65	CO > 0.65
K1	1.091	0.809	1.091	.809
K2	-0.948	-0.891	-0.948	-0.891
K3	887.46	387.53	887.46	387.53
K4	0.726	0.587	0.726	0.587
K5	0.333	0.096	0.0	0.0
K6	0.186	0.203	0.0	0.0

$$(CS.25) \qquad V_{cx}^{n+1} = \left\{\frac{d_{co}}{d_{ci}h'_{ci}} + R_{fc} + \frac{d_{co}\ \ell n\ (d_{co}/d_{ci})}{2\ K_{mc}[T_p]}*\frac{1}{12} + \frac{1}{h_{co}}\right\}^{-1}$$

$$(CS.26) \qquad Q_{\Delta x}^{n+1} = \pi d_{ci}\left(\left(\frac{U_{cx}^{n+1} - U_{prev}}{2}\right)\Delta x + U_{prev}\ \Delta x\right) * \frac{1}{12}$$

$$(CS.27) \qquad \chi^{n+1} = \frac{Q_{\Delta x}^{n+1} + (Q_{\Delta x})\ SUM}{h_{Rfg}\ [T_p]}$$

$$(CS.28) \qquad I = \frac{L_{REM}}{3n}\left\{U_{cx}\Big|_{x=0} + U_{cx}\Big|_{x=L_{REM}} + 4\sum_{\substack{j=1 \\ odd}}^{n-1} U_j + 2\sum_{\substack{j=1 \\ even}}^{n-2} U_j\right\}$$

(CS.29) $$I = \frac{x}{2\Delta x}\left\{U_{cx}\Big|_{x\ 0} + U_{cx}\Big|_{x} + 2\sum_{j=1}^{n-1} U_j\right\}$$

(CS.30) $$h_p = h_{Rv}\left[T_p, v_{R\ell}\left[T_p\right]\right] - h_{Rfg}\left[T_p\right]$$

(CS.31) $$Re_{ci}' = \frac{4\ M_{SK}}{\Pi\ d_{ci}\ \mu_{w\ell}\left[T_p\right]\ N_c}$$

(CS.32) $$K_f = \frac{J\ h_{wfg}\left[T_p\right]\ (\chi_x - \chi_o)}{g\qquad x}$$

(CS.33) $$f_{Tc}' = 0.0185\ \left(\frac{K_f}{Re_{ci}'}\right)^{0.25}$$

(CS.34) $$\chi_m = 0.5\ (\chi_x + \chi_o)$$

(CS.35) $$v_m = v_{w\ell}\left[T_p\right] + \chi_m\ \left(v_{wv}\left[T_p\right] - v_{w\ell}\left[T_p\right]\right)$$

(CS.36) $$\Delta P_{TC}'' = \left(\frac{f_{TC}'\ x}{d_{ci}} * 12 \div \frac{(\chi_x - \chi_o)}{\chi_m}\right)\ G^2 \cdot v_m$$

(CS.37) $$T_q = T_{WSAT}\left[P_{WSAT}\left[T_p\right] - \Delta P_{TC}''\right]$$

(CS.38) $$h_q = \chi_x\ h_{Rfg}\left[T_q\right]$$

If the steam experiences superheat, equations (C.1) to (C.39) must be re-executed with the following alterations,

- $$Re_{ci} = \frac{4\ M_{SK}}{\Pi\ d_{ci}\mu_{wv}\left[T_q\right]} * 12$$

- (C.5) $$h_{ci} = \frac{K_{wv}\left[T_q\right]}{d_{ci}}\ (0.023)Re_{ci}^{0.8}\ \left\{\frac{C_{pwv}\left|T_q\right|\mu_{wv}\left[T_q\right]}{K_{wv}\left[T_q\right]}\right\}^{0.36} * 12$$

$$\bullet \quad \text{(C.6)} \quad h_{ci} = \frac{K_{wv}[T_q]}{d_{ci}} \; (3.66) * 12$$

$$\text{(CS.39)} \quad Q'''_{CDS} = (M_{SK} - M_{SC} - M_{DSH})\, C_{pwv}\, \varepsilon_c\, (CT - T_q)$$

$$\text{(CS.40)} \quad Q_{CD} = (M_{SK} - M_{SC} - M_{DSH})\, C_{pw\ell}[0.5(T_p + T_7)](T_p - T_7) + I + Q'''_{CDS}$$

$$\text{(CS.41)} \quad Q_C = Q_{CD} + Q_{SC} + Q_{DSH}$$

EXPANSION VALVE

Coefficients A_{ox} and A_{1x} in equation (EX.1) must be obtained externally since they are derived from performance data correlations.

$$\text{(EX.1)} \quad M_X = \frac{1}{v_{R\ell}[T_{RSAT}[P_{RSAT}[ET] + \Delta P_6]]} * 60\ (.13368) * 10^{C_1}$$

$$\text{Where} \quad C_1 = \frac{\log\,(P_5 - P_6) - A_{ox}}{A_{1X}}$$

5.5 LOGIC FLOW-CHARTING

This section is provided as a visual aid in structuring the equations presented in section 5.2.3, as explained throughout section Four.

In the interest of minimizing flow-chart complexity, certain conventions were adopted and their meanings are described below:

* TOL

  this term is used with loop structures and identifies the exit condition as being the iterative tolerance of the identified variable and /or equation.

* "/"

  the slask mark denotes concurrently executed statements.

* n

  this letter represents the current iteration of a given variable.

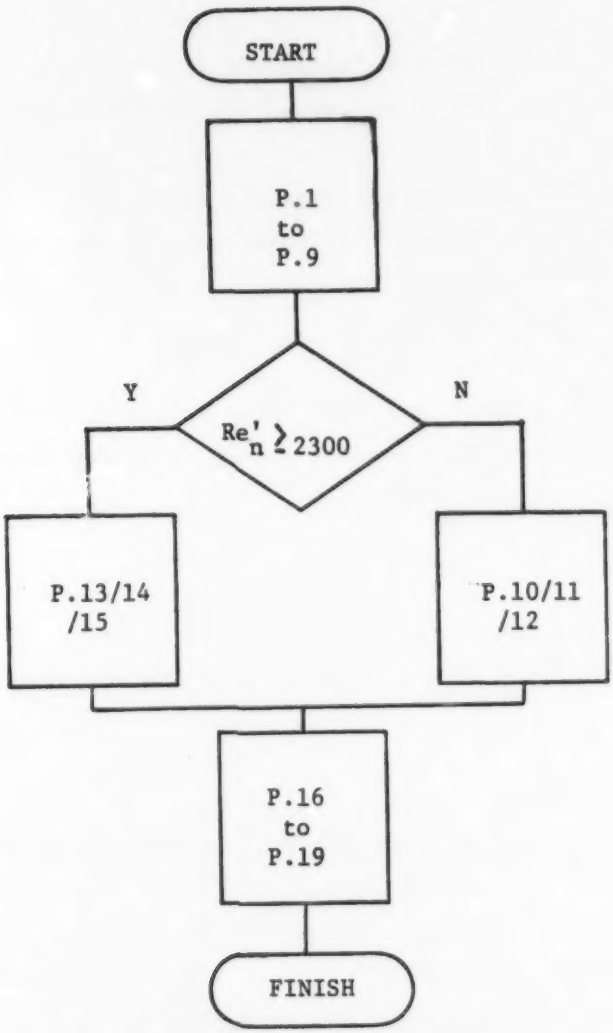
* n+1

  this combination represents the next iteration of a given variable.

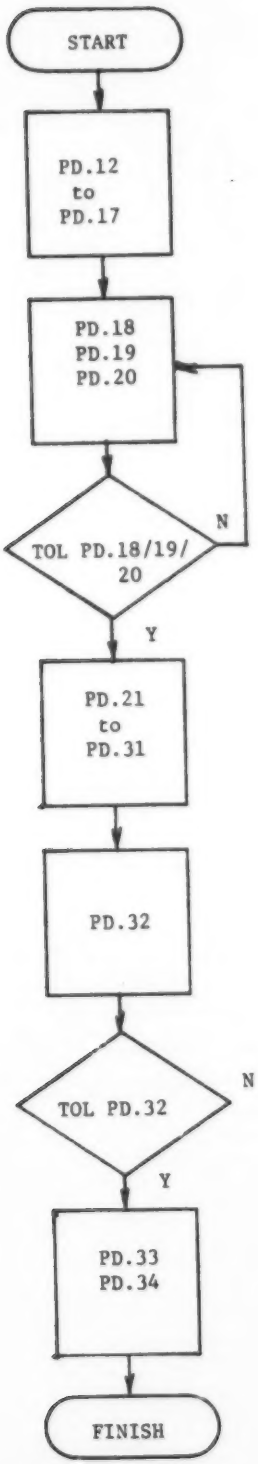
* prev

  this denotes the previous value of a given variable.

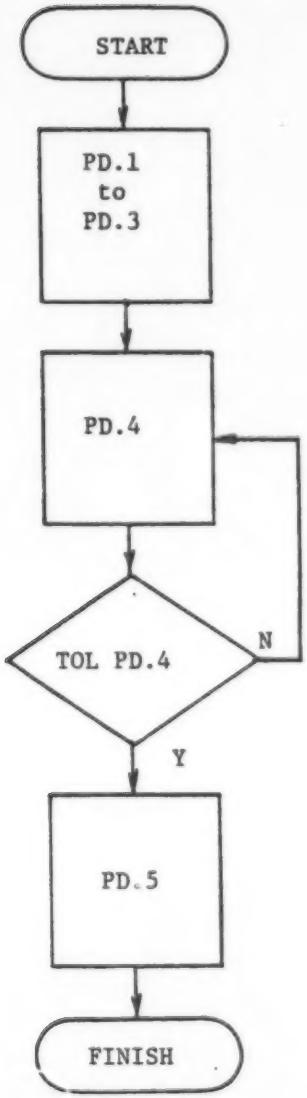
PIPE DESIGN


START
P.1
to
P.9
Y
$Re'_n \geq 2300$
N
P.13/14
/15
P.10/11
/12
P.16
to
P.19
FINISH

PRESSURE DROP
PIPING SYSTEM

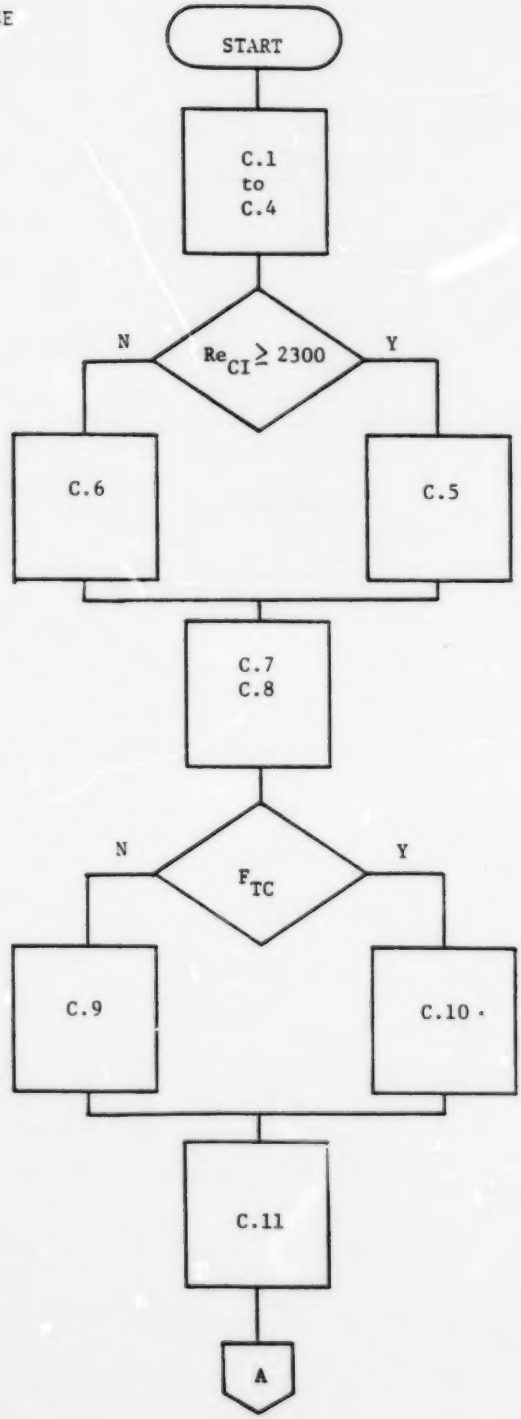


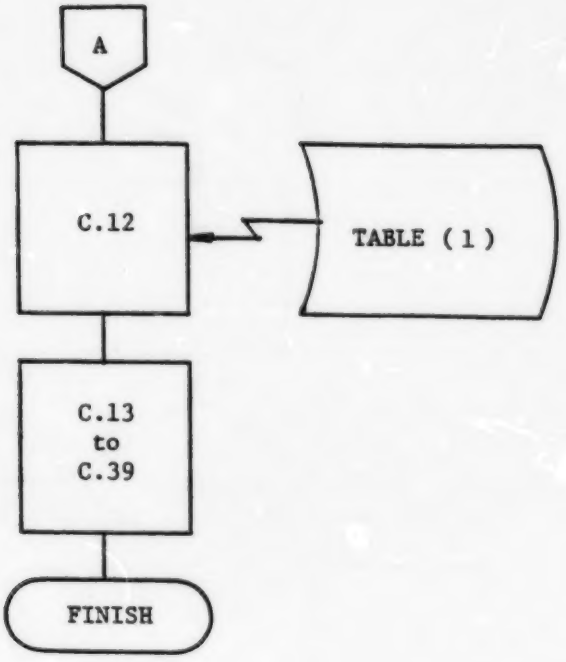
PRESSURE DROP
EVAPORATOR
COOLANT



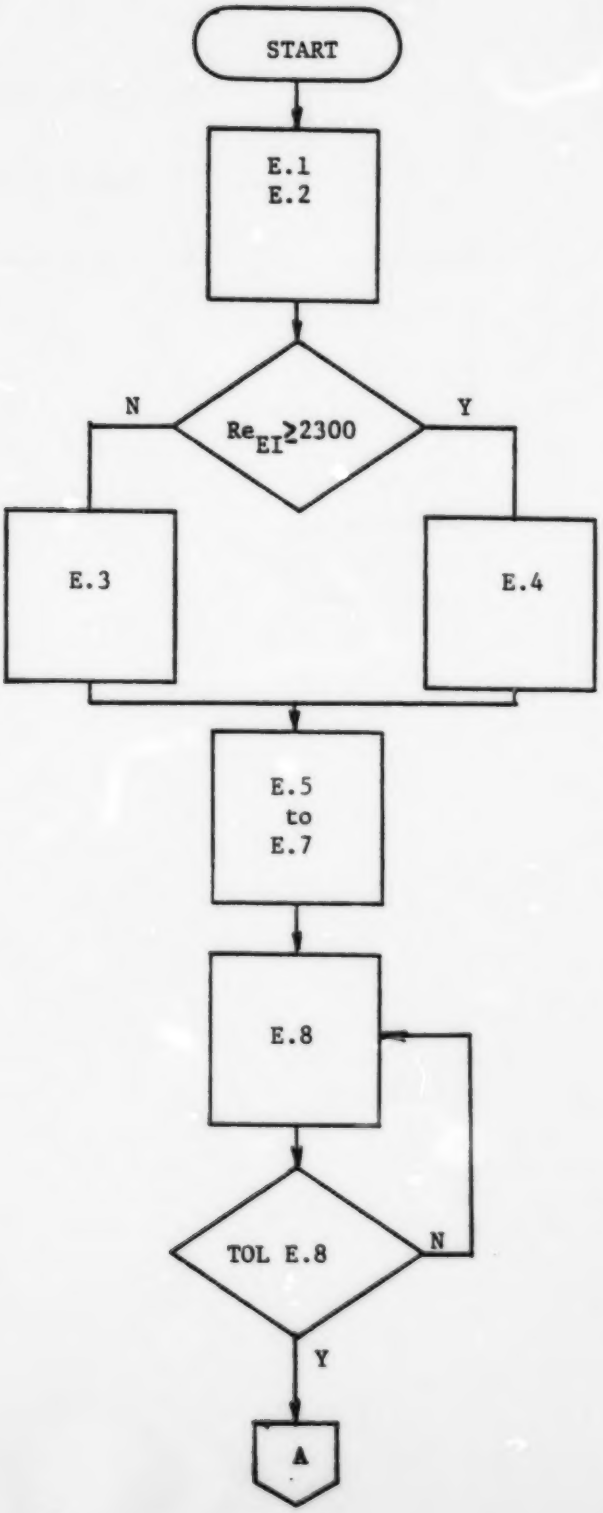
STATEPOINT ALTERATIONS

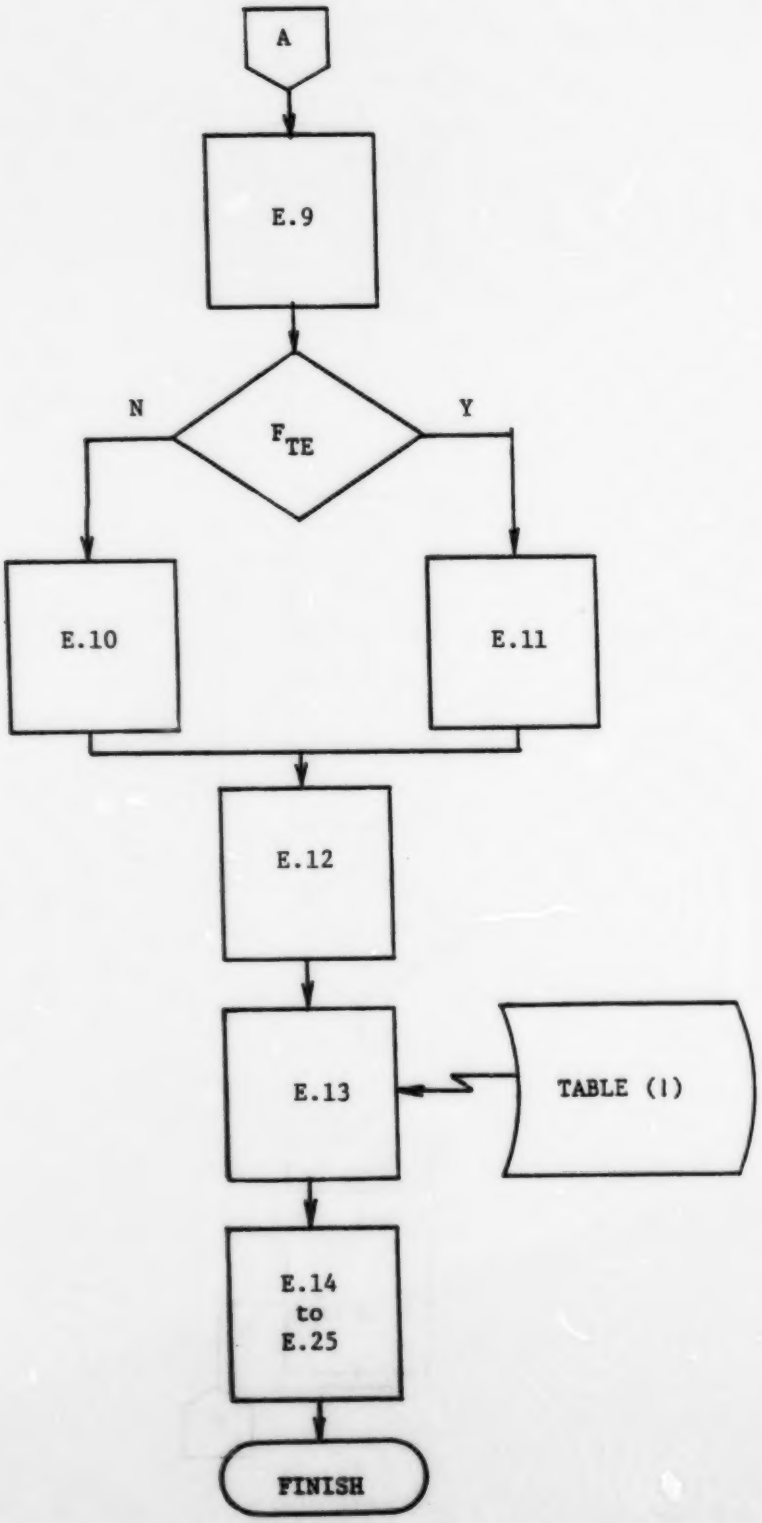
CONDENSER
SINGLE-PHASE
COOLANT



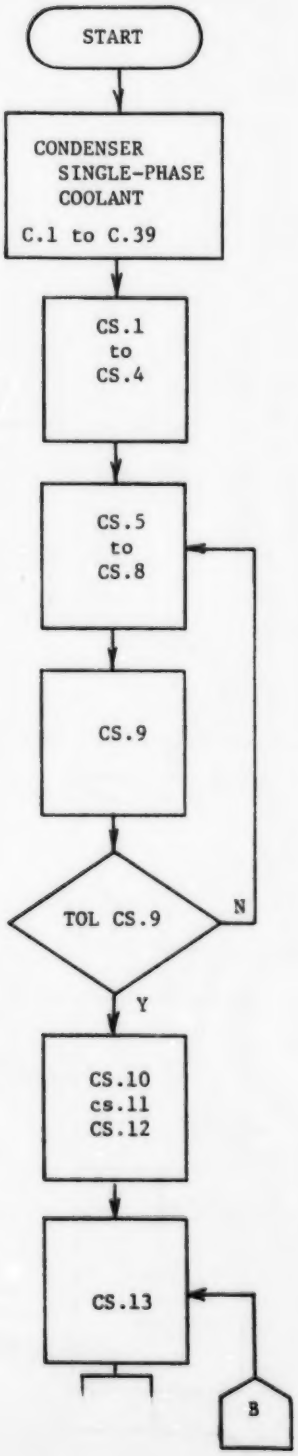

A
C.12
TABLE (1)
C.13
to
C.39
FINISH

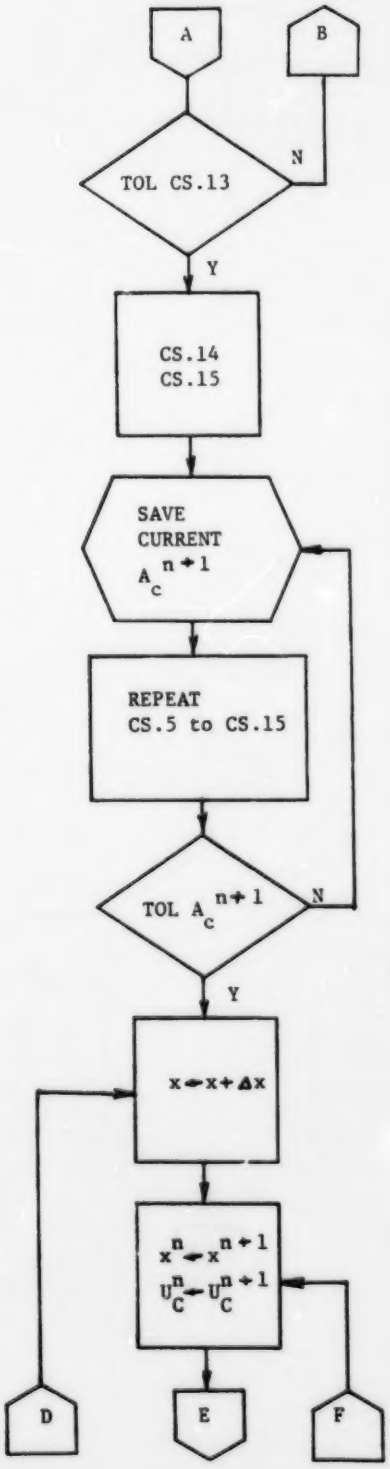
EVAPORATOR

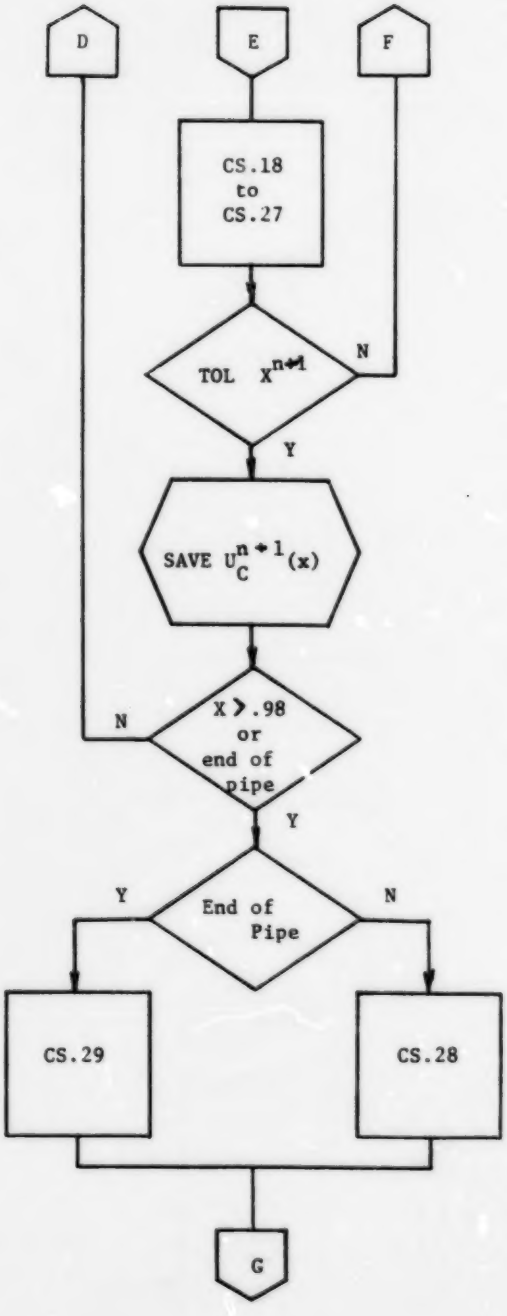


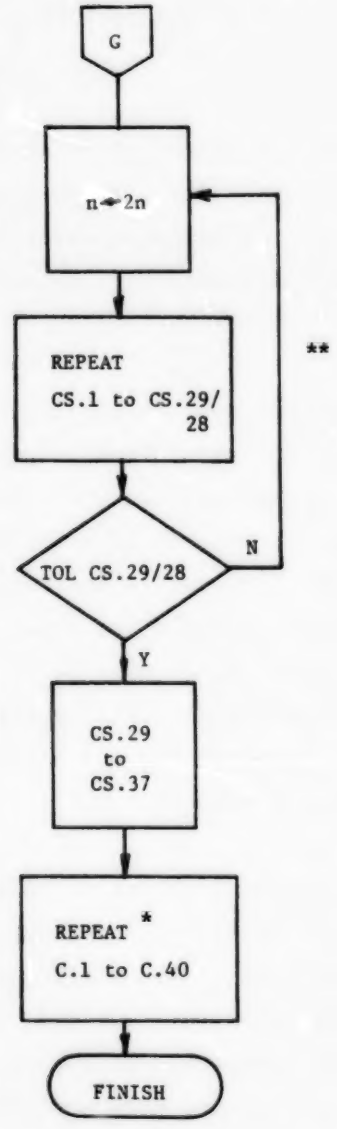

A
E.9
N
F_{TE}
Y
E.10
E.11
E.12
E.13
TABLE (1)
E.14
to
E.25
FINISH

CONDENSER
TWO-PHASE COOLANT




A
B
TOL CS.13
N
Y
CS.14
CS.15
SAVE
CURRENT
A_c^{n+1}
REPEAT
CS.5 to CS.15
TOL A_c^{n+1}
N
Y
$x \leftarrow x + \Delta x$
$x^n \leftarrow x^{n+1}$
$U_C^n \leftarrow U_C^{n+1}$
D
E
F


D
E
F
CS.18
to
CS.27
TOL X^{n+1}
N
Y
SAVE $U_C^{n+1}(x)$
N
X > .98
or
end of
pipe
Y
Y
End of
Pipe
N
CS.29
CS.28
G



* use same equations and procedure as before except substitute c_{pwv} for c_{pwl}

** start at x = 0 (i.e. liquid saturation point) assuming X = 0 at x = 0 and $U_C^n = U_C$ as provided by the condenser/single-phase coolant equations (C.1 to C.39)

6.0 LIMITATIONS

This section discusses the restrictions and perceived weaknesses in the proposed simulation. Essentially, it is a comment on the usefulness of the system as a whole. The critique involves the following areas of concern: convergence, configuration, formulae and assumptions, testing.

Identification of the heat pump refrigerant cycle is accomplished through the use of a non-linear simultaneous equations solver. However, there is no guarantee that the obtained solution will be physically realistic. It depends upon the nature of the equations used and the initial estimate provided. In fact, the possibility of a divergent solution exists. Since the relationships used realistically model the various components, it must be assumed that a region of convergence exists for a solution vector reasonably close to the actual solution. The extent to which the estimated solution vector can vary from the real solution is unknown.

Another convergence concern involves the possibility of the solver wandering into regions of operation for which either component or property data does not exist. The difficulty, which is true of the version of Brown's system solver provided, is that the region for which a solution is searched cannot be restricted. Although unavailable at the time of writing, certain methods such as the "Nedler-Mead Simplex Solver" exist which do possess this feature.

A third convergence concern is with regard to the weak effect which the quantity of superheat has on the final solution. Because the amount of heat transfer ensuing from superheat is minor in comparison with the latent heat of vaporization, convergence may occur even with an unreasonable initial estimate of superheat. Hence, realistic estimates should always be furnished for superheat under the assumption that it may not change.

The "generic" heat pump approach to modelling provides great flexibility in the type of heat pump configuration which can be simulated. The major configuration restriction limits coolant flow to the tubeside of both heat exchangers modelled. Note that the coolant is not limited to water but evaporator coolant flow must be single-phase. Recall also that knowledge of inter-component piping is not a requirement. Note that the compressor is not limited to the screw type since the component is modelled on the basis of performance data. Provided that another compressor can provide the required information, interchangeability is acceptable.

Another topic of concern includes the actual assumptions or correlations employed in the modelling. Throughout the simulation, components were generally assumed to contribute no heat loss. Relatively speaking, this is generally the case. However, if heat loss were a problem, it is assumed that the devices can be insulated. Only the compressor oil heat rejection rate is formally considered. With regard to the correlations used, some may prove questionable. In determing the external convection coefficient for the heat exchanger tube bundles, correlations were employed which assumed an existing cross-flow situation as occurs for baffled heat exchangers. Generally, of course, evaporators and condensers are unbaffled. Also, the evaporator model does not account for the contribution of tube fins to the outside tube convection coefficient directly. The equations however doubtful, are assumed adequate. Nevertheless, revisions may be required pending the outcome of real system prediction.

Finally, it must be realized the current presented simulation has not trully been tested in a fully integrated form. Although some modules were tested with fictitious data and produced reasonable results of the proper magnitude, without real data to substantiate results, these tests are of no consequence. A priority of subsequent follow up work must be to compare predicted results of this simulation with data for a specified configuration. Sufficient detail of the components should exist to run the simulation (i.e. number of tubes, tube sizes, fin details, etc.). Because of the inability to run a system test, values

which constitute a reasonable tolerance for convergence purposes were unattainable. These tolerances are required for subsequent tests in order to identify an acceptable degree of convergence. This is important since unreasonable or unattainable tolerances would either prolong or prevent convergence on a solution.

CONCLUSIONS

The present work describes the simulation of a vapour compression heat pump optionally with steam generating capability. The system has been designed as an eventual computer application.

The intent of this simulation is to predict heat pump operation in sufficient detail to compare individual unit performance. In this manner, a heat pump for any given application can be optimized on the basis of source or sink conditions and overall capacity. To accomplish this, the simulation requires wide applicability with many vapour compression heat pumps. This requirement is satisfied by reducing each heat pump to a rudimentary generic unit. The various components of the generic heat pump are modelled using fundamental thermodynamic heat transfer equations and detail performance specifications.

Due to the unavailability of real industrial heat pump data, this proposed simulation has not been validated. This is left as a task for a subsequent continuation of this work.

RECOMMENDATIONS

The simulation presented should be validated by comparison with data obtained from heat pumps in industrial applications. Discrepancies which occur should be sourced and appropriate modifications to the component models undertaken.

In addition, as a result of the validation procedure tolerances which describe a reasonable degree of convergence should be established for subsequent simulation usage.

In order to reduce execution time, performance and property data should be stored directly in memory as opposed to files. An alternative approach is to adopt accurate correlations to describe data curves.

If the validation procedure demonstrates the problem of solution estimates regularly wandering into unmapped data regions, the non-linear simultaneous equation system solver should be substituted. As a replacement, a scheme restricting the region of convergence should be adopted.

ACKNOWLEDGEMENTS

I would like to recognize the assistance of Jim Wright, Jeff Linton and Richard Varotta without whose insight and experience this work would not have been possible.

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APPENIDX A

Refrigerant Equations

by R.C. Dowing

No. 2313

REFRIGERANT EQUATIONS

R. C. DOWNING
Fellow ASHRAE

A knowledge of the thermodynamic properties of refrigerants is essential for comparing actual with theoretical performance, for estimating the desirability of using a refrigerant in a given application, and for comparing one refrigerant with another. In the past, equations have been developed to describe thermodynamic behavior and used to calculate tables of properties. The tables were then used to calculate refrigeration cycle data, and the equations, having served their purpose, were no longer of general interest.

Now, with computers readily available, the thermodynamic properties of a refrigerant and the derived data of interest in a refrigeration cycle can be calculated easily and quickly for the conditions desired. Lengthy and laborious hand calculations, often necessary when using tables that seldom included data at the right temperature and pressure conditions, can be eliminated. With a computer, equations are of direct interest.

Methods of developing equations and their use in calculating thermodynamic data have been thoroughly discussed in the literature. This subject will not be discussed here except to mention a few references. A complete review of correlations and equations used in calculating thermodynamic properties was published by Martin (1). Later, Schofield (2) restated the thermodynamic relationships and listed the results of some computer calculations for various refrigeration cycles. Kartsounes and Erth (3) published a complete computer program for calculating properties of Refrigerants 12, 22, and 502.

The equations used in calculating tables of thermodynamic properties for refrigerants have appeared in different publications or, in some cases, have not been published at all. In view of the renewed interest in equations, it seems desirable to assemble as many as possible in one place. Equations seem to be more desirable than computer programs, since computer language is subject to change and users may prefer different formats.

Objectives

The objectives of this report are:

1. To assemble the constants for several refrigerants that can be used in general forms of the equations necessary for the calculation of thermodynamic and refrigeration properties.
2. To illustrate how the English equations can be used to generate data in metric units.
3. To show that some simplification of the equation of state is possible without affecting the calculation of properties in the region of interest in refrigeration and air conditioning.

General Equations

Equations for the following four basic properties of refrigerants are used in calculating thermodynamic relationships:

1. Liquid density
2. Vapor pressure
3. Pressure-volume-temperature relationships of the vapor
4. Heat capacity of the vapor

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The experimental data for any property can be represented in different ways. In most cases, the general forms of the equations presented here were developed by Prof. Martin at the University of Michigan.

1. LIQUID DENSITY

$$d_L = A_L + B_L\left(1-\frac{T}{T_c}\right)^{1/3} + C_L\left(1-\frac{T}{T_c}\right)^{2/3} + D_L\left(1-\frac{T}{T_c}\right) + E_L\left(1-\frac{T}{T_c}\right)^{4/3} + F_L\left(1-\frac{T}{T_c}\right)^{1/2} + G_L\left(1-\frac{T}{T_c}\right)^{2} \quad (1)$$

d_L = Lbs/cuft

T = F + 459.67 (459.69, 459.6, 459.7 in some cases)

T_c = Critical temperature in R

The constants for a number of refrigerants are listed in Table 1.

TABLE 1. CONSTANTS FOR LIQUID DENSITY EQUATION

Refrigerant	A_L	B_L	C_L	D_L	E_L	F_L	G_L	T_C R	Add to F	Ref.
11	34.57	57.63811	43.6322	−42.82356	36.70663	0	0	848.07	459.67	4
12	34.84	53.341187	0	18.69137	0	21.98396	−3.150994	693.3	459.7	5
13	36.06996	54.395124	0	8.512776	0	25.879906	9.589006	543.60	459.69	6
14	39.06	69.568489	4.5866114	36.1716662	−8.058986	0	0	409.50		7
21*	116.37962	−0.03106808	−0.0000501	0	0	0	0	812.9	459.6	8
22	32.76	54.634409	36.74892	−22.2925657	20.4732886	0	0	664.50	459.69	9
23	32.7758	63.37784	−25.30533	144.16182	−106.1328	0	0	538.33	459.69	10
113*	122.872	−0.0128	0.0000636	0	0	0	0	877.0	459.6	8
114	36.32	61.146414	0	16.418015	0	17.476838	1.119828	753.95	459.69	11
500	31.00	43.562	74.709	−87.583	56.483	0	0	681.59	459.69	12
502	35.0	53.48437	63.86417	−70.08066	48.47901	0	0	639.56	459.67	13
C318	38.70	70.858318	23.609759	15.989182	−8.8243856	0	0	699.27	459.69	14

*The form of the equation is $d_L = A_L + B_L T + C_L T^2$

2. VAPOR PRESSURE

$$\log_{10} P = A + \frac{B}{T} + C\log_{10} T + DT + E\left(\frac{F-T}{T}\right)\log_{10}(F-T) \quad (2)$$

P = psia

T = F + 459.67 (459.69, 459.6, 459.7 in some cases)

The constants for several refrigerants are listed in Table 2

3. EQUATION OF STATE

$$P = \frac{RT}{V-b} + \frac{A_2 + B_2T + C_2e^{-KT/T_c}}{(V-b)^2} + \frac{A_3 + B_3T + C_3e^{-KT/T_c}}{(V-b)^3}$$

$$+ \frac{A_4 + B_4T + C_4e^{-KT/T_c}}{(V-b)^4} + \frac{A_5 + B_5T + C_5e^{KT/T_c}}{(V-b)^5}$$

$$+ \frac{A_6 + B_6T + C_6e^{-KT/T_c}}{e^{\alpha V}(1 + C'e^{\alpha V})} \qquad (3)$$

P = psia

V = Cuft/lb

T = F + 459.67 (459.69, 459.6, 459.7 in some cases)

The constants for several refrigerants are given in Table 3.

TABLE 2. CONSTANTS FOR VAPOR PRESSURE EQUATION

Refrigerant	A	B	C	D	E	F	Add to F	Ref.
11	42.14702865	−4344.343807	−12.84596753	4.0083725×10^{-3}	0.0313805356	862.07	459.67	4
12	39.88381727	−3436.632228	−12.47152228	4.73044244×10^{-3}	0	0	459.7	5
13	25.967975	−2709.538217	−7.17234391	2.545154×10^{-3}	0.280301091	546.00	459.67	15
14	20.71545389	−2467.505285	−4.69017025	6.4798076×10^{-4}	0.770707795	424	459.69	16
21	42.7908	−4261.34	−13.0295	3.9851×10^{-3}	0	0	459.6	17
22	29.35754453	−3845.193152	−7.86103122	2.1909390×10^{-3}	0.445746703	686.1	459.69	9
23*	328.90853	−7952.76913	−144.5142304	0.24211502	-2.1280665×10^{-4}	9.434955×10^{-8}	459.69	10
113	33.0655	−4330.98	−9.2635	2.0539×10^{-3}	0	0	459.6	17
114	27.071306	−5113.7021	−6.3086761	6.913003×10^{-4}	0.78142111	768.35	459.69	11
500	17.780935	−3422.69717	−3.63691	5.0272207×10^{-4}	0.4629401	695.57	459.67	18
502	10.644955	−3671.153813	−0.369835	-1.746352×10^{-3}	0.8161139	864	459.67	13
C318	15.63242	−4301.063	−2.128401	-1.19759×10^{-3}	0.6625896	714	459.69	14

*Form of equation is: $\log_{10}P = A + \frac{B}{T} + C\log_{10}T + DT + ET^2 + FT^3$

TABLE 3. CONSTANTS FOR EQUATION OF STATE

	REFRIGERANT			
	11	12	13	14
R	0.078117	0.088734	0.102728	0.1219336
b	0.00190	0.0065093886	0.0048	0.0015
A_2	−3.126759	−3.40972713	−3.083417	−2.162959
B_2	1.318523×10^{-3}	1.59434848×10^{-3}	2.341695×10^{-3}	2.135114×10^{-3}
C_2	−35.76999	−56.7627671	−18.212643	−18.941131
A_3	−0.025341	0.0602394465	0.058854	4.404057×10^{-3}
B_3	4.875121×10^{-5}	-1.87961843×10^{-5}	-5.671268×10^{-5}	1.282818×10^{-5}
C_3	1.220367	1.31139908	0.571958	0.539776
A_4	1.687277×10^{-3}	-5.4873701×10^{-4}	-1.026061×10^{-3}	1.921072×10^{-4}
B_4	-1.805062×10^{-6}	0	1.338679×10^{-6}	-3.918263×10^{-7}
C_4	0	0	0	0
A_5	-2.358930×10^{-5}	0	5.290649×10^{-6}	-4.481049×10^{-6}
B_5	2.448303×10^{-8}	3.468834×10^{-9}	-7.395111×10^{-9}	9.062318×10^{-9}
C_5	-1.478379×10^{-4}	-2.54390678×10^{-5}	-3.874233×10^{-5}	-4.836678×10^{-5}
A_6	1.057504×10^{8}	0	7.378601×10^{7}	5.838823×10^{7}
B_6	-9.472103×10^{4}	0	-7.435565×10^{4}	-9.263923×10^{4}
C_6	0	0	0	0
K	4.50	5.475	4.00	4.00
α	580	0	625	661.199997
c'	0	0	0	0
T_C R	848.07	693.3	543.60	409.50
Add to F	459.67	459.7	459.67	459.69
Ref.	4	5	16	16

Note: When α is zero, any term in which it appears in the denominator should be omitted.

	REFRIGERANT			
	21	22	23	113
R	0.10427	0.124098	0.15327	0.05728
b	0	0.002	0.00125	0
A_2	−7.316	−4.353547	−4.679499	−4.035
B_2	4.6421×10^{-3}	2.407252×10^{-3}	3.472778×10^{-3}	2.618×10^{-3}
C_2	0	−44.066868	−159.775232	0
A_3	−0.20382376	−0.017464	0.012475	−0.0214
B_3	3.593×10^{-4}	7.62789×10^{-5}	7.733388×10^{-5}	5.00×10^{-5}
C_3	0	1.483763	5.941212	0
A_4	0	2.310142×10^{-3}	2.068042×10^{-3}	0
B_4	0	-3.605723×10^{-6}	-3.684238×10^{-6}	0
C_4	0	0	0	0
A_5	0	-3.724044×10^{-5}	-3.868548×10^{-5}	0
B_5	0	5.355465×10^{-8}	6.455643×10^{-8}	0
C_5	0	-1.845051×10^{-4}	-7.394214×10^{-4}	0
A_6	0	1.363387×10^{8}	7.502357×10^{7}	0
B_6	0	-1.672612×10^{5}	-1.114202×10^{5}	0
C_6	0	0	0	0
K	0	4.2	5.50	0
α	0	548.2	520.0	0
c'	0	0	0	0
$T_{C'}$, F	812.9	664.50	538.33	877.0
Add to F	459.69	459.69	459.69	459.69
Ref.	19	9	16	19

	REFRIGERANT			
	114	500	502	C-318
R	0.062780807	0.10805000	0.096125	0.053646698
b	0.005914907	0.006034229	0.00167	0.0060114165
A_2	−2.3856704	−4.549888	−3.2613344	−1.8947274
B_2	1.0801207×10^{-3}	2.308415×10^{-3}	2.0576287×10^{-3}	9.8484745×10^{-4}
C_2	−6.5643648	−92.90748	−24.24879	−28.542156
A_3	0.034055687	0.08660634	0.034866748	0.026479892
B_3	-5.3336494×10^{-6}	-3.141665×10^{-5}	-8.6791313×10^{-6}	-6.862101×10^{-6}
C_3	0.16366057	2.742282	0.33274779	0.66384636
A_4	-3.857481×10^{-6}	-8.726016×10^{-4}	-8.5765677×10^{-4}	-2.4565234×10^{-4}
B_4	0	0	7.0240549×10^{-7}	0
C_4	0	0	0.022412368	0
A_5	1.6017659×10^{-6}	-1.375958×10^{-6}	8.8368967×10^{-6}	6.0887086×10^{-7}
B_5	6.2632341×10^{-10}	9.149570×10^{-9}	-7.9168095×10^{-9}	8.269634×10^{-10}
C_5	-1.0165314×10^{-5}	-2.102661×10^{-4}	-3.7167231×10^{-4}	-3.849145×10^{-5}
A_6	0	0	-3.8257766×10^{7}	0
B_6	0	0	5.5816094×10^{4}	0
C_6	0	0	1.5378377×10^{9}	0
K	3.0	5.475	4.2	5
a	0	0	609	0
c'	0	0	7×10^{-7}	0
$T_{C'}$ R	753.95	681.59	639.56	699.27
Add to F	459.69	459.69	459.67	459.69
Ref.	11	12	13	14

4. HEAT CAPACITY OF THE VAPOR

$$C_v = a + bT + cT^2 + dT^3 + \frac{f}{T^2} - \frac{JK^2T\,e^{-KT/T_c}}{T_c^2}\left[\frac{C_2}{V-b}\right. \qquad (4)$$

$$+ \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} + \frac{C_5}{4(V-b)^4} + \frac{C_6}{\alpha\alpha^{\alpha v}}$$

$$\left. - \frac{C_6 C'}{\alpha}\,(\ell\eta 10)\log_{10}\left(1 + \frac{1}{C'e^{\alpha v}}\right)\right]$$

C_v = Heat capacity at constant volume = Btu/(lb) (R)

T = F + 459.67 (459.69, 459.6, 459.7 in some cases)

J = 0.185053

$\ell\eta 10$ = Natural logarithm of 10 = 2.30258509

Some constants for a number of refrigerants are listed in Table 4.
The other constants in the above equation are from the equation of state.

TABLE 4. CONSTANTS FOR HEAT CAPACITY EQUATION

Refrigerant	a	b	c	d	f	Add to F	Ref.
11	0.023815	2.798823×10^{-4}	-2.123734×10^{-7}	5.999018×10^{-11}	-336.80703	459.67	4
12	8.0945×10^{-3}	3.32862×10^{-4}	-2.413896×10^{-7}	6.72363×10^{-11}	0	459.7	5
13	0.01602	2.823×10^{-4}	-1.159×10^{-7}	0	0	459.69	6
14	0.0300559282	2.3704335×10^{-4}	-2.85660077×10^{-8}	$-2.95338805\times10^{-11}$	0		7
21	0.0427	1.40×10^{-4}	0	0	0	459.6	20
22	0.02812836	2.255408×10^{-4}	-6.509607×10^{-8}	0	257.341	459.69	9
23	0.07628087	-7.561805×10^{-6}	3.9065696×10^{-7}	-2.454905×10^{-10}	0	459.69	10
113	0.07963	1.159×10^{-4}	0	0	0	459.6	20
114	0.0175	3.49×10^{-4}	-1.67×10^{-7}	0	0	459.69	11
500	0.026803537	2.8373408×10^{-4}	-9.7167893×10^{-8}	0	0	459.69	21
502	0.020419	2.996802×10^{-4}	-1.409043×10^{-7}	2.210861×10^{-11}	0	459.67	13
C318	0.0225178157	3.69907814×10^{-4}	-1.64842522×10^{-7}	$2.152780846\times10^{-11}$	0	459.69	14

DERIVED EQUATIONS

The four basic equations can be combined by exact thermodynamic relationships to calculate properties used in refrigeration and air conditioning. Some of these relationships are shown below:

1. LATENT HEAT OF VAPORIZATION

$$\Delta H_{lat} = JT\left(V_g - V_f\right)\left[P\,\ell\eta 10\left(\frac{-B}{T^2} + \frac{C}{T\ell\eta 10} + D - E\left(\frac{\log_{10}e}{T} + \frac{F\log_{10}(F-T)}{T^2}\right)\right)\right] \quad (5)$$

ΔH_{lat} = Latent heat of vaporization, Btu/lb.

J = 0.185053

V_g = Saturated volume of vapor, cuft/lb.

V_f = Saturated volume of liquid, cuft/lb.

P = Vapor pressure, psia

$\ell\eta 10$ = Natural logarithm of 10 = 2.302585093

T = R

$\log_{10}e$ = 0.4342944819

Use constants from the vapor pressure equation.

2. ENTHALPY OF VAPOR

$$H = aT + \frac{bT^2}{2} + \frac{cT^3}{3} + \frac{dT^4}{4} - \frac{f}{T} + JPV + J\left[\frac{A_2}{V-b} + \frac{A_3}{2(V-b)^2} + \frac{A_4}{3(V-b)^3} + \frac{A_5}{4(V-b)^4} + \frac{A_6}{a}\left(\frac{1}{e^{av}} - C'\left[\ell\eta 10\right]\log\left[1 + \frac{1}{C'e^{av}}\right]\right)\right] + Je^{-KT/T_c}\left(1 + \frac{KT}{T_c}\right)\left[\frac{C_2}{V-b} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} + \frac{C_5}{4(V-b)^4} + \frac{C_6}{ae^{av}} - \frac{C_6C'(\ell\eta 10)\log}{a}\left(1 + \frac{1}{C'e^{av}}\right)\right] + X \quad (6)$$

3. ENTROPY OF THE VAPOR

$$S = a(\ell\eta 10)\log T + bT + \frac{cT^2}{2} + \frac{dT^3}{3} - \frac{f}{2T^2} + JR(\ell\eta 10)\log(V-b) - J\left[\frac{B_2}{V-b} + \frac{B_3}{2(V-b)^2} + \frac{B_4}{3(V-b)^3} + \frac{B_5}{4(V-b)^4} + \frac{B_6}{a}\left(\frac{1}{e^{av}} - C'\left[\ell\eta 10\right]\log\left[1 + \frac{1}{C'e^{av}}\right]\right)\right] + \frac{JKe^{-KT/T_c}}{T_c}\left[\frac{C_2}{V-b} + \frac{C_3}{2(V-b)^2} + \frac{C_4}{3(V-b)^3} + \frac{C_5}{4(V-b)^4} + \frac{C_6}{ae^{av}} - \frac{C_6C'(\ell\eta 10)\log}{a}\left(1 + \frac{1}{C'e^{av}}\right)\right] + Y \quad (7)$$

NOTES FOR SECTIONS 3 & 5

S = Btu/(lb)(R)

H = Btu/lb

T = R

J = 0.185053

P = Pressure, psia

V = Volume, cuft/lb

a,b,c,d,f = Constants from heat capacity equation

A,B,C,K,α,c' = Constants from the equation of state

X = ΔH(latent at reference temperature)-H(vapor at reference temperature)

$$Y = \frac{\Delta H(\text{latent at reference temperature})}{T\ (\text{reference})} - S(\text{vapor at reference temperature})$$

In most cases the enthalpy and entropy of the liquid are assumed to be zero at -40 F.

With Refrigerant 14, the heat capacity of the solid and liquid was estimated from absolute zero and this reference point was used.

Values for constants X and Y are listed in Table 5.

TABLE 5. CONSTANTS X AND Y FOR THE ENTHALPY AND ENTROPY EQUATIONS

Refrigerant	X	Y
11	50.5418	-0.0918395
12	39.556551	-0.016537936
13	20.911	-0.05676
14	86.102162	0.36172528
21	-	-
22	62.4009	-0.0453335
23	-	-
113	25.198	-0.40552
114	25.3398621	-0.11513718
500	48.4734	-0.09012707564
502	35.308	-0.07444
C318	12.19214242	-0.16828871

METRIC UNITS

Thermodynamic properties in metric units can be calculated in two different ways.

1. Convert the constants in the basic equations to give the desired units.
2. Use the equations with English constants and convert the result.

The first method is perhaps the better one and such conversions may be made in the future. In the meantime, the second method can be used to generate metric data if desired. Some conversion factors are illustrated in Table 6 for metric units commonly used in refrigeration and air conditioning. The only complication is in shifting the reference point for enthalpy and entropy. Common practice in the metric system is to assign a value of 100 kcal/kg for the enthalpy of the liquid at 0 C and a value of 1.0 kcal/(kg)(K) for the entropy of the liquid at 0 C. Commonly used reference points in SI units are 200 kjoule/kg for the enthalpy of the liquid at 0 C and 1.0 kjoule/(kg)(K) for the entropy of the liquid at 0 C.

TABLE 6. METRIC CONVERSION FACTORS

Property	English Unit	Operation	Factor	Metric Unit
Temperature	R	Divide	1.8	K
Pressure	Psia	Multiply	0.070306958	kg/cm^2
		Multiply	0.068046	atmospheres
		Multiply	0.0689476	bars
Volume	cuft/lb	Multiply	62.4261	liters/kg
		Multiply	0.062427961	$(meters)^3/kg$
		Multiply	62.427961	$(decimeters)^3/kg$
Density	lbs./cuft	Multiply	0.01601891	kg/liter
Enthalpy (latent)	Btu/lb	Divide	1.8	kcal/kg
		Multiply	4.1868	kjoules/kg
			1.8	
Entropy (latent)	Btu/(lb)(R)	Multiply	1	kcal/(kg)(K)
		Multiply	4.1868	kjoules/(kg)(K)

Enthalpy (vapor):

$$H \text{ in kcal/kg} = \frac{H_T + \Delta H_{32} - H_{32}}{1.8} + 100$$

$$H \text{ in kj/kg} = \frac{4.1868}{1.8}\left[H_T + \Delta H_{32} - H_{32}\right] + 200$$

Entropy (vapor):

$$S \text{ in kcal/(kg)(K)} = S_T + \Delta S_{32} - S_{32} + 1$$

$$S \text{ in kjoule/(kg)(K)} = 4.1868\left[S_T + \Delta S_{32} - S_{32}\right] + 1$$

H_T = Enthalpy of the vapor at any temperature and pressure in Btu/lb

ΔH_{32} = Enthalpy of vaporization at 32 F in Btu/lb

H_{32} = Enthalpy of saturated vapor at 32 F in Btu/lb

S_T = Entropy of the vapor at any temperature and pressure in Btu/(lb)(R)

ΔS_{32} = Entropy of vaporization at 32 F in Btu/(lb)(R)

S_{32} = Entropy of saturated vapor at 32 F in Btu/(lb)(R)

EQUATION OF STATE

Most of the equations listed here and the experimental data on which they are based were developed by Professor J. J. Martin and coworkers at the University of Michigan. The principal interest there has been to modify the equation of state (the Martin-Hou equation) to represent the data at high densities. In order to reach higher densities, the equation has become increasingly complex.

For applications in refrigeration and air conditioning it is not necessary to use the full equation. The 6th term of the equation,

$$\frac{A_6 + B_6T + C_6e^{-\frac{KT}{T_c}}}{e^{\alpha v}(1 + C'e^{\alpha v})}$$

can be omitted without any sacrifice in accuracy. A comparison of the pressure calculated with and without the 6th term for R22 and R502 is shown in Table 7.

If the equation of state is used without the 6th term, these constants should also be omitted wherever they appear in derived equations.

Table 7 -- Effect of Omitting the 6th Term in the Equation of State for Refrigerants R22 and R502.

Percent difference in pressure omitting the 6th term compared to the full equation.

	R-22				R-502		
Specific Volume, ⟶ cu ft/lb (Saturation temp., ⟶ F)	0.02	0.03	0.04 (203)	0.05 (199)	0.02	0.03 (179+)	0.04 (178)
180 F					15	0.0007	0
200	60	0.27	0.0011	0	10	0.0005	0
220	36	0.19	0.0008	0	7	0.0004	0
240	24	0.14	0.0006	0	5	0.0003	0
260	16	0.10	0.0005	0	4	0.0003	0
280	10	0.07	0.0003	0	3	0.0002	0

Critical Specific Volume:

R22, .0305 cu ft/lb
R502, .0896 cu ft/lb

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APPENDIX B

Specifications for a Typical Industrial Heat Pump

DESIGN SPECIFICATIONS FOR R114/STEAM COMPRESSOR HEAT PUMP

OBJECTIVES

The objectives of this project are to demonstrate steam generation using a newsprint mill white water source, R114 refrigerant, oil injected screw compressors in a steam generating heat pump and the integration of the heat pump with a steam compressor to supply steam at up to 240 kpa (35 psig) suitable for newsprint paper machine dryer cylinders. Participation of Canadians in technical aspects is important. Maximum Canadian content for design, equipment, material and labour will be required.

EQUIPMENT SPECIFICATIONS

1. Refrigerant Evaporator - a flooded R114 refrigerant evaporator with suction accumulator is to be supplied. The heat source for the evaporator will be ground wood white water, the white water will be on the tube side of the shell and tube evaporator and all areas of contact with white water should be 316SS. White water is water which has been removed from wood pulp, a fibre/water suspension produced in a pulp mill and used to make newsprint, it contains approximately 1% fibre.

   The thermal properties, viscosity, hydrodynamic characteristics and heat transfer properties of white water can be assumed to be the same as those of water. There can be problems with fouling and tube plugging unless the velocity is kept above 1.7 m/sec (6 feet per second). The final report of the Pulp and Paper Research Institute of Canada (M.K. Azarniouch) and Consolidated Bathurst (Y. Lemay) on experiments carried out for the Federal Department of Energy, Mines and Resources deals at length with this subject. This report will be provided to the successful bidders.

   The flow of white water should have as few restrictions as possible and the heads should be removable for cleaning. The velocity of the white water should be not less than 2.5 m/sec. (8 ft/sec.).

   The design requirements of the evaporator are as follows:

Heat removal	=	10GJ/HR (9,480,000 BTU/HR)
White water flow rate	=	10,000 Litre/Min. (2642 USGPM)
White water inlet temp.	=	68°C (155°F)

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2. Condenser/Steam Generator - an R114 condenser/steam generator will be required with R114 in the tubes or shell (whichever design proves more efficient).

   The design requirements of the condenser are as follows:

   Total heat rejection = to match input of evaporator and compressor.

   Inlet feedwater temp. = 65.6°C (150°F)

   Outlet steam condition = 68 kpa (10 psig) saturated steam.

3. R114 Liquid Subcooler - provide an R114 liquid subcooler between the condenser liquid outlet and evaporator inlet. The subcooler suction should be connected to a side load port of the refrigerant screw compressor.

4. R114 Suction/Liquid Heat Exchanger - to ensure that the suction temperature is high enough to prevent the condensing of R114 during compression in the screw compressor.

5. R114 High Pressure Receiver - provide enough storage to hold the entire refrigerant charge.

6. Oil Return System - provide an oil return system to remove oil from the R114 flooded evaporator and return to the compressor.

7. Refrigerant Screw Compressor - provide a high temperature screw compressor suitable for R114 at the temperatures expected in the system c/w the following.
   - High efficiency oil separator c/w manway for access to elements.
   - Dual oil filters with one being standby.
   - Remote oil pump.
   - Suction side port for liquid subcooler.
   - Automatic capacity reduction from 10% to 100%.

8. Electric Motors - all large electric motors will be 2200 volt/3/60 and smaller motors will be 575/3/60 all motors should have a 1.15 service factor.
   The compressor drive motors will be protected by thermistors in each winding and have class "F" insulation.

9. Motor Control Centre - a separate motor control centre (not package mounted) shall be provided with individual

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enclosures and fused disconnects for each motor starter. A 110 volt control transformer will be supplied for package control wiring. The panels will be NEMA 12 dust-tight enclosure.

10. Steam Compressor - provide a centrifugal type single stage steam compressor c/w capacity control modulation.

The design requirements of the steam compressor are as follows:

Mass flowrate = to match maximum output of steam generator.

Inlet steam press. = 68KPA (10psig).

Outlet steam press. = 240KPA (35psig).

As an alternate to the above centrifugal steam compressor a water flooded screw steam compressor could be investigated. (Please note that the system with a centrifugal compressor must be quoted as the base quotation).

If a screw steam compressor is used the output of the condenser/steam generator can be lowered to a minimum of 34KPA (5psig) if it improves efficiency, because of the steam screw compressors ability to handle higher pressure ratios.

The proposed screw steam compressor must be a proven design with demonstrated trouble free operation.

The report should contain a discussion of the advantages and disadvantages of the screw type steam compressor compared with the base case single stage centrifugal compressor.

Steam screw compressors are manufactured by ThermoElectron, Atlas-Copco and Mycom.

11. Control System - the control of all equipment on the package will be handled by a programable micro-processor system housed in a NEMA 12 dust-tight enclosure.

All required safety controls shall be included to ensure that any fault will not damage equipment.

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All required control devices shall be included with the package.

Pressure and temperature transducers shall be located at all strategic locations on the package and connected to a digital readout at the control panel.

12. Package Description - all the above mentioned equipment shall be mounted of a rigid steel base with adequate bracing to prevent vibration transmission to floor. The package size shall allow for ease of transportation and rigging on site and may consist of more than one skid, with allowances for field connection between skids.

The package shall be completely wired, insulated and all inter-connecting refrigerant and steam piping is to be provided.

The package is to be primed and finish painted after assembly.

All piping and vessels are to be in accordance with the CSA B52 code.

13. Refrigerant and Oil Charge - a complete charge of R114 and synthetic refrigerant oil is to be included with the package care should be exercised to ensure the most compatible type of oil is selected to match the refrigerant and the temperatures expected.

NOTE: The heat pump is expected to be operating on a continuous year-round basis, all equipment should be selected accordingly.

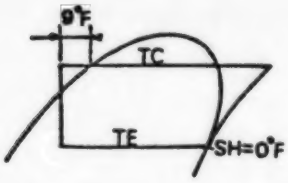
APPENDIX C

Typical Screw Compressor Data

Symbols in these capacity tables are as in the following.

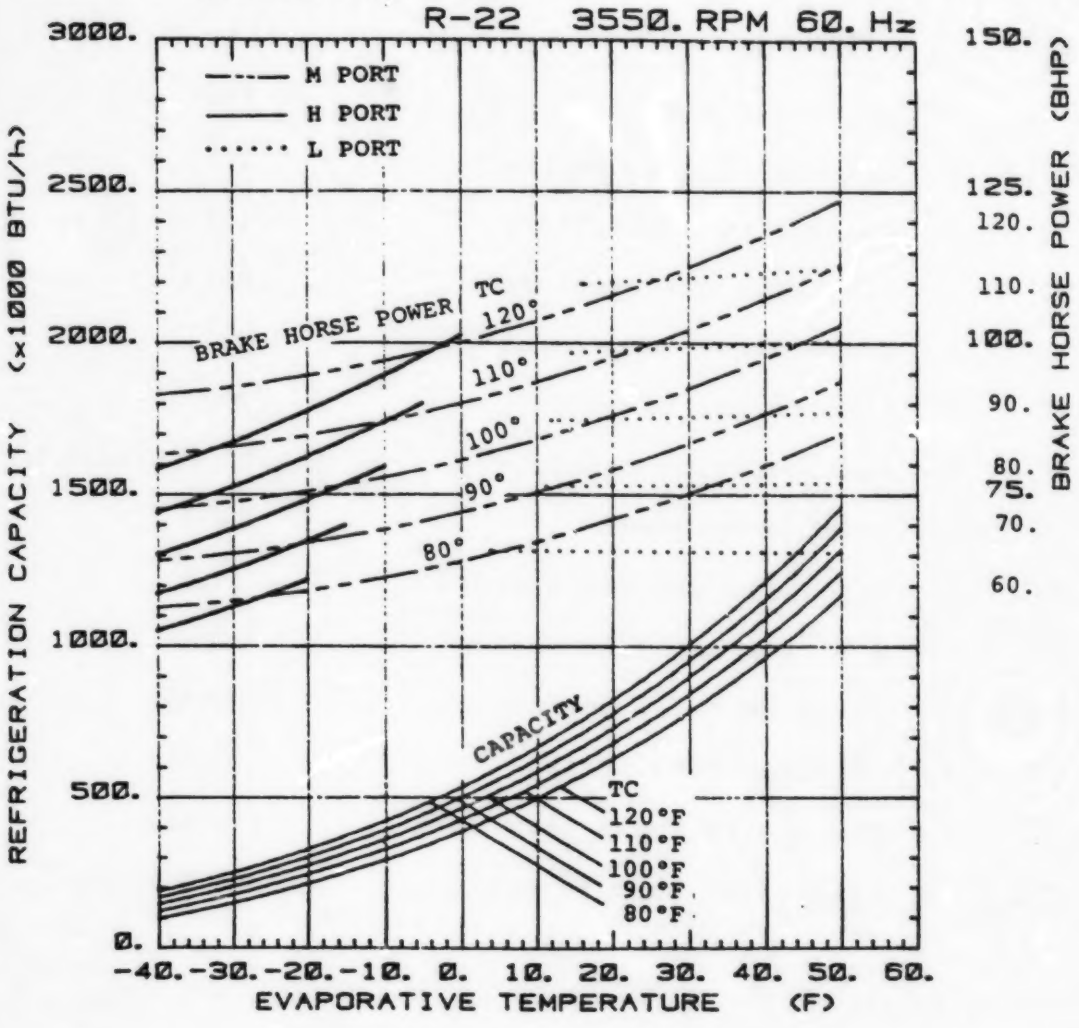
VTH	:	Theoretical displacement	(CFM)
SH	:	Degree of suction superheat	(°F)
SC	:	Degree of liquid subcooling	(°F)
Toil	:	Oil supply temperature	(°F)
TC	:	Condensing temperature	(°F)
PD	:	Discharge gas pressure	(PSIA)
TE	:	Evaporative temperature	(°F)
PS	:	Suction gas pressure	(PSIA)
RT	:	Refrigeration capacity	(x1000 BTU/hr)
KW	:	Break horsepower	(HP)
TD	:	Discharge gas temperature	(°F)
QI	:	Oil flow rate, injection	(GPM)
QJ	:	Oil flow rate, journal	(GPM)
QT	:	Oil flow rate, total	(GPM)
GL	:	Refrigerant mass flow rate	(lb/hr)
OHR	:	Oil cooler heat rejection	(x1000 BTH/hr)

Technical information from Mayekawa Mfg. Co. Ltd., TD-JS06, Aug., 1981



F125LUD(LG)2-M

R-22 3550. RPM 60. Hz



[C A U T I O N]

CURVES MUST NOT BE EXTRAPOLATED.

Technical information from Mayekawa Mfg. Co. Ltd., TD-ASA6-05, Aug., 1981

F125LU2- M 3550. RPM R22
VTH : 208. CFM
S H : 0. deg.
S C : 9. deg.
Toil : 104. F

**** TC : 120.0 F / PD : 274.6 PSIA ****

TE (F)	PS (PSIA)	RT (BTU/h) ×1000.	BHP (BHP)	TD (F)	QI (GPM)	QJ (GPM)	QT (GPM)	GL (lb/h)	OHR (BTU/h) ×1000.
50.	98.7	1169.	124.	159.	2.2	6.4	8.6	17656.	96.
40.	83.2	962.	118.	158.	3.4	6.5	9.9	14713.	109.
30.	69.6	784.	112.	158.	4.1	6.6	10.8	12139.	118.
20.	57.7	630.	108.	159.	4.7	6.7	11.4	9895.	127.
10.	47.5	499.	104.	160.	5.1	6.8	11.9	7949.	136.
0.	38.7	387.	100.	162.	5.4	6.9	12.3	6269.	145.
-10.	31.2	293.	97.	165.	5.7	6.9	12.6	4827.	155.
-20.	24.8	215.	95.	168.	5.9	7.0	12.9	3597.	166.
-30.	19.6	150.	93.	171.	6.1	7.0	13.1	2554.	179.
-40.	15.2	96.	91.	175.	6.2	7.1	13.3	1676.	192.

**** TC : 110.0 F / PD : 241.0 PSIA ****

TE (F)	PS (PSIA)	RT (BTU/h) ×1000.	BHP (BHP)	TD (F)	QI (GPM)	QJ (GPM)	QT (GPM)	GL (lb/h)	OHR (BTU/h) ×1000.
50.	98.7	1246.	113.	152.	1.2	5.8	7.0	17950.	68.
40.	83.2	1029.	107.	151.	2.5	6.0	8.5	15008.	81.
30.	69.6	842.	102.	151.	3.5	6.1	9.5	12435.	90.
20.	57.7	681.	98.	151.	4.1	6.2	10.3	10194.	99.
10.	47.5	544.	94.	153.	4.6	6.2	10.8	8250.	107.
0.	38.7	427.	90.	155.	4.9	6.3	11.2	6573.	116.
-10.	31.2	328.	87.	157.	5.2	6.4	11.6	5134.	125.
-20.	24.8	246.	85.	161.	5.5	6.4	11.9	3908.	136.
-30.	19.6	177.	83.	164.	5.6	6.4	12.1	2870.	147.
-40.	15.2	121.	82.	168.	5.8	6.5	12.3	1997.	160.

**** TC : 100.0 F / PD : 210.6 PSIA ****

TE (F)	PS (PSIA)	RT (BTU/h) ×1000.	BHP (BHP)	TD (F)	QI (GPM)	QJ (GPM)	QT (GPM)	GL (lb/h)	OHR (BTU/h) ×1000.
50.	98.7	1320.	103.	144.	.4	5.3	5.7	18197.	47.
40.	83.2	1094.	97.	144.	1.4	5.4	6.9	15256.	56.
30.	69.6	899.	92.	143.	2.7	5.5	8.3	12684.	66.
20.	57.7	730.	88.	144.	3.5	5.6	9.1	10444.	74.
10.	47.5	587.	84.	145.	4.1	5.7	9.8	8503.	82.
0.	38.7	465.	81.	147.	4.5	5.8	10.2	6828.	90.
-10.	31.2	361.	78.	150.	4.8	5.8	10.6	5393.	99.
-20.	24.8	275.	76.	153.	5.0	5.9	10.9	4170.	109.
-30.	19.6	204.	74.	157.	5.2	5.9	11.2	3136.	119.
-40.	15.2	145.	73.	161.	5.4	6.0	11.3	2267.	131.

Technical information from Mayekawa Mfg. Co. Ltd., TD-ASA6-05, Aug., 1981

F125LU2- M 3550. RPM R22
VTH : 208. CFM
S H : 0. deg.
S C : 9. deg.
Toil : 104. F

**** TC : 90.0 F / PD : 183.1 PSIA ****

TE (F)	PS (PSIA)	RT (BTU/h) ×1000.	BHP (BHP)	TD (F)	QI (GPM)	QJ (GPM)	QT (GPM)	GL (lb/h)	OHR (BTU/h) ×1000.
50.	98.7	1391.	94.	136.	0.0	4.8	4.8	18403.	31.
40.	83.2	1156.	88.	136.	.5	5.0	5.5	15463.	36.
30.	69.6	953.	84.	136.	1.7	5.1	6.7	12892.	44.
20.	57.7	778.	79.	136.	2.9	5.2	8.0	10653.	53.
10.	47.5	628.	75.	138.	3.5	5.2	8.7	8714.	60.
0.	38.7	501.	72.	140.	4.0	5.3	9.3	7041.	67.
-10.	31.2	393.	69.	142.	4.3	5.4	9.7	5608.	75.
-20.	24.8	303.	67.	146.	4.6	5.4	10.0	4388.	84.
-30.	19.6	228.	65.	149.	4.8	5.4	10.3	3357.	94.
-40.	15.2	167.	64.	154.	5.0	5.5	10.5	2492.	105.

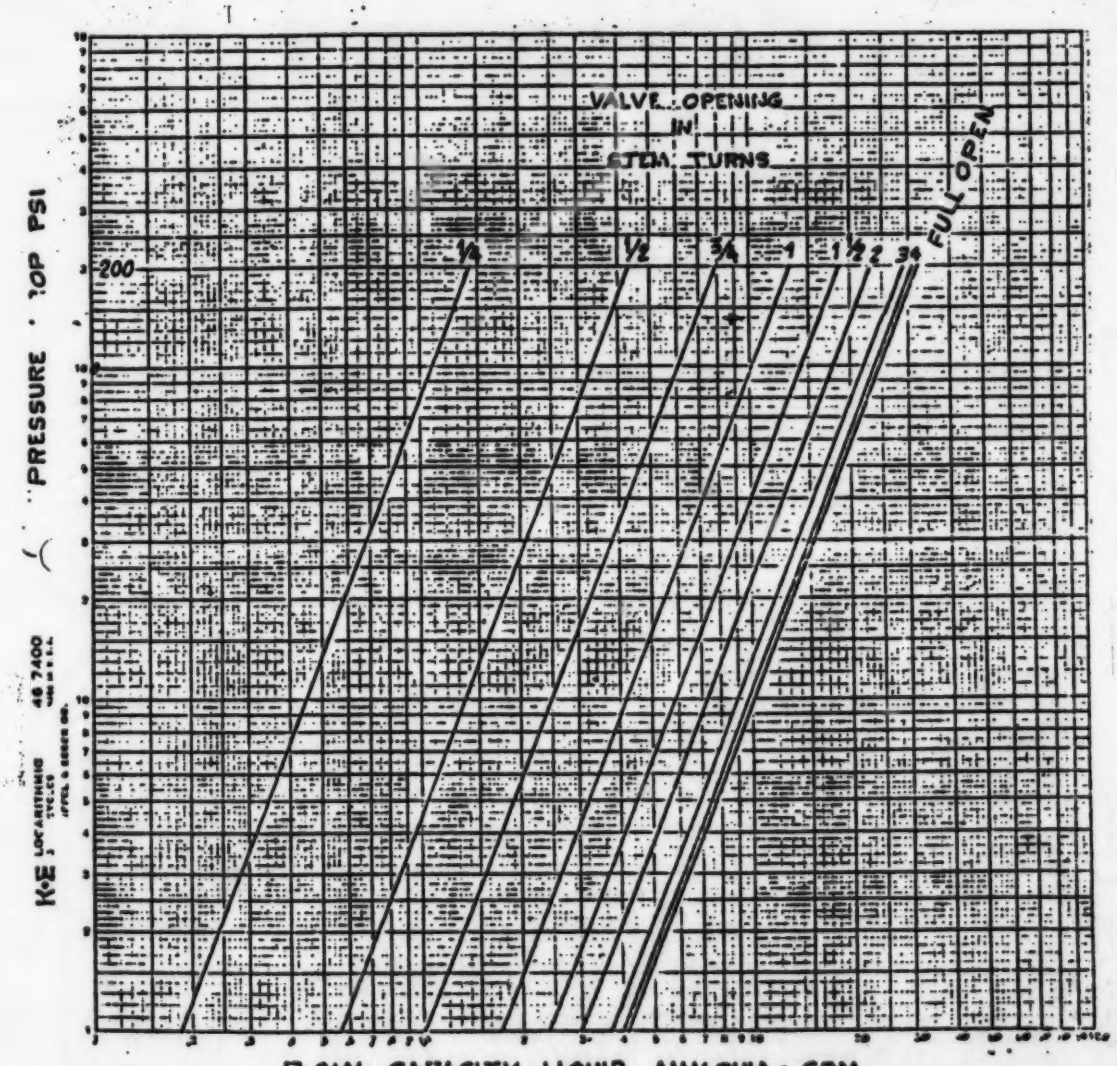
**** TC : 80.0 F / PD : 158.3 PSIA ****

TE (F)	PS (PSIA)	RT (BTU/h) ×1000.	BHP (BHP)	TD (F)	QI (GPM)	QJ (GPM)	QT (GPM)	GL (lb/h)	OHR (BTU/h) ×1000.
50.	98.7	1459.	85.	127.	0.0	4.4	4.4	18572.	20.
40.	83.2	1216.	80.	128.	0.0	4.5	4.5	15632.	22.
30.	69.6	1005.	75.	129.	.9	4.6	5.5	13063.	27.
20.	57.7	823.	71.	129.	2.1	4.7	6.8	10825.	34.
10.	47.5	667.	67.	130.	2.9	4.8	7.8	8887.	41.
0.	38.7	535.	64.	132.	3.5	4.9	8.4	7216.	48.
-10.	31.2	423.	61.	135.	3.9	4.9	8.8	5785.	55.
-20.	24.8	329.	59.	138.	4.2	5.0	9.2	4568.	63.
-30.	19.6	251.	57.	142.	4.4	5.0	9.5	3539.	72.
-40.	15.2	187.	56.	146.	4.6	5.1	9.7	2678.	82.

Technical information from Mayekawa Mfg. Co. Ltd., TD-JSA6-05', Aug., 1981

APPENDIX D

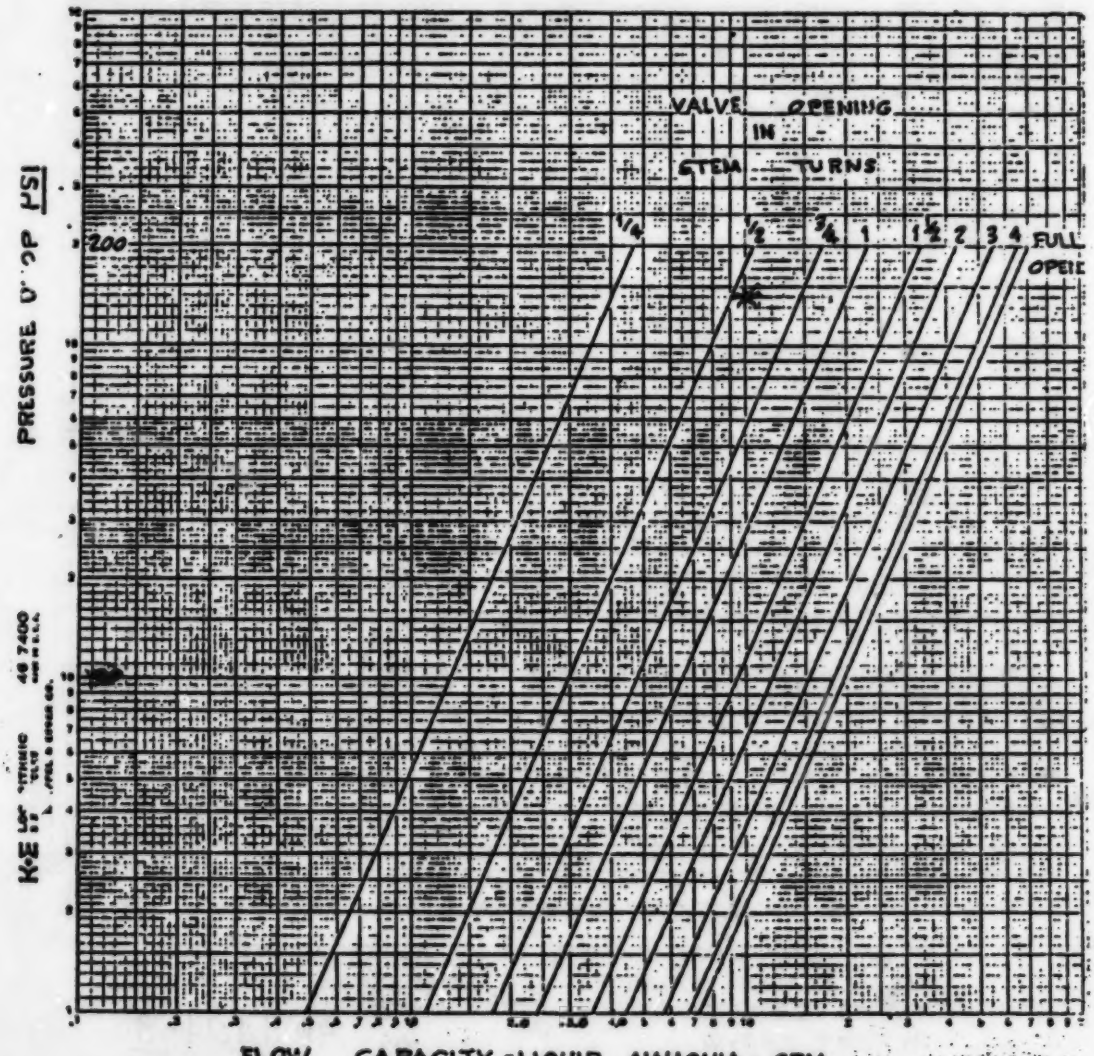
Expansion Valve Data



FLOW CAPACITY CHART
3/4 METERING VALVE
CAT. NO. 530A-E
REPORT 288-W

HENRY VALVE CO.
MELROSE PARK, ILL
DWG. NO. AE 1404-3

Flow Rate ASHRAE 32.27



FLOW CAPACITY CHART
1" METERING VALVE
CAT. NO. 340 A-E
REPORT 289W

HENRY VALVE CO
MELROSE PARK, ILL
DWG.
NO. AE 1405-3
MGT.

APPENDIX E

A version of Brown's Non-linear Simultaneous

```
      SUBROUTINE ZSYSTM (F,EPS,NSIG,N,X,ITMAX,WA,EVAL,IER)
C
C      ----- DETERMINATION OF A ROOT OF A SYSTEM OF N
C      ----- SIMULTANEOUS NONLINEAR EQUATIONS IN N
C      ----- UNKNOWN, F(X)=0, IN VECTOR FORM.
C     F     - F IS THE NAME OF THE DOUBLE PRECISION
C             FUNCTION CALLED BY ZSYSTM TO FURNISH THE
C             VALUES OF THE FUNCTIONS WHICH DEFINE THE
C             SYSTEM OF EQUATIONS BEING SOLVED. THE USER
C     EPS   - FIRST STOPPING CRITERION. A ROOT X(1), ... ,
C             X(N) IS ACCEPTED IF THE MAXIMUM ABSOLUTE
C             VALUE OF F(X,K,PAR) IS LESS THAN OR EQUAL
C             TO EPS, WHERE K=1, ... ,N. (INPUT)
C     NSIG  - SECOND STOPPING CRITERION. A ROOT IS ACCEPTED
C             IF TWO SUCCESSIVE APPROXIMATIONS TO A GIVEN
C             ROOT AGREE IN THE FIRST NSIG DIGITS. (INPUT)
C             NOTE: IF EITHER, OR BOTH, OF THE STOPPING
C             CRITERIA ARE SATISFIED, THE ROOT IS ACCEPTED.
C     N     - THE NUMBER OF EQUATIONS (= NUMBER OF UNKNOWNS)
C             (INPUT)
C     X     - THE VECTOR X OF LENGTH N, AS INPUT, IS THE
C             INITIAL GUESS FO THE ROOT.  AS OUTPUT, IT
C             IS THE COMPUTED SOLUTION.
C     ITMAX- ON INPUT = THE MAXIMUM ALLOWABLE NUMBER OF
C             ITERATIONS AND ON OUTPUT = YHE NUMBER OF
C             ITERATIONS USED IN FINDING THE COMPUTED
C             SOLUTION.
C     WA    - AN ARRAY WORK AREA OF SIZE ((N+2)*(N-1))/2 +
C             3*N
C     EVAL  - SUBROUTINE EVAL (X,F) CONTAINS THE FUNCTION
C             F(X)=0, WHICH DEFINES THE SYSTEM OF EQUATIONS
C     IER   - ERROR PARAMETER
C             TERMINAL ERROR = 128 +N
C             N = 1 INDICATED FAILURE TO CONVERGE WITHIN
C                   ITMAX ITERATIONS.
C             N = 2 INDICATED SINGULARITY (OF THE JACOBIAN
C                   MATRIX) HAS BEEN ENCOUNTED TWICE.
C
      REAL X(50),F(50),WA(1500)
      REAL*8 PREC,DELTA,PM1,PT1,P2
      DATA PREC,DELTA /5.D-14,5.D-10/
      DATA PM1,PT1,P2 /.1D0,.0001D0,.002D0/
      IER=0
      ITMAX=MIN0(ITMAX,50)
      N2=N+N
      RELCON=10.D+0**(-NSIG)
      JTEST = 1
      ZERO=0.0
      IERROR=0
      IPART=((N+2)*(N-1))/2
      ITMP=IPART+N
      LKSUB=ITMP+N
      DO 130 M = 1, ITMAX
         IQUIT=0
         FMAX=0.0
```

```
      M1 = M-1
      K1 = LKSUB + 1
      KMIN = LKSUB+N
      XTEMP =0.0
      IF (ITMAX.EQ.0) GOTO 1003
      CALL EVAL (X,F)
      WRITE (6,1002) M1
 1002 FORMAT (2X,'ITERATION',I3)
      WRITE (6,1000) (X(LL),LL=1,N)
      WRITE (6,1001) (F(LL),LL=1,N)
 1000 FORMAT (2X,'SOLUTION VECTOR',2X,9D13.4)
 1001 FORMAT (2X,'RESIDUAL',9X,9D13.4)
C
C                THE ARRAY WA(LKSUB+1),...,WA(LKSUD+N)
C                PERMITS A PARTIAL PIVITING EFFECT
C                WITHOUT THE PHYSICAL INTERCHANG OF
C                ROWS OR COLUMNS.
C
 1003 DO 5 J=K1,KMIN
         XTEMP=XTEMP+1.DO
         WA(J)=XTEMP
    5 CONTINUE
      K=1
   10 IF (K.LE.1) GOTO 30
      KMIN=K-1
C
C                THE FOLLOWING CODE BACK-SOLVES THE
C                FIRST KMIN ROWS OF A TRIANGULARIZED
C                LINEAR SYSTEM FOR IMPROVED X VALUES
C                IN TERMS OF PREVIOUS ONES.
C
      KK=1
   15 DO 25 K1=1,KMIN
         ISUB=K-K1
         MM=((ISUB-1)*(N2-ISUB))/2
         LIM=N-ISUB
         KPOINT=WA(LKSUB+ISUB)+PM1
         ISUB1=ISUB-1
         X(KPOINT)=ZERO
         DO 20 L1=1,LIM
            JS1=ISUB1+L1
            LKJSUB=LKSUB+JS1+1
            IJ=MM+JS1
            JPOINT=WA(LKJSUB)+PM1
            X(KPOINT)=X(KPOINT)+WA(IJ)*X(JPOINT)
   20       CONTINUE
            X(KPOINT)=X(KPOINT)+WA(MM+N)
   25 CONTINUE
      GOTO (30,45,105), KK
C
C                SET UP PARTIAL DERIVATIVES OF
C                KTH FUNCTION.
C
   30 CONTINUE
```

```
      CALL EVAL (X,F)
      E=F(K)
      FMAX=AMAX1(FMAX,ABS(E))
      IF(ABS(E).GE.EPS) GOTO 35
      IQUIT=IQUIT+1
      IF(IQUIT.EQ.N) GOTO 140
   35 I=K
   40 IP=IPART+I
      ITEMP=WA(LKSUB+I) + PM1
      HOLD = X(ITEMP)
      ETA=.001D+00*ABS(HOLD)
      IF (ABS(HOLD) .LT.PREC)  ETA=DELTA
      H=AMIN1(FMAX,ETA)
      IF(H.LT.PREC) H=PREC
      X (ITEMP)=HOLD+H
      IF(K.LE.1) GOTO 45
      KK=2
      GOTO 15
   45 CONTINUE
      CALL EVAL (X,F)
      FPLUS = F(K)
      TOP=FPLUS-E
      IF(ABS(TOP).GE.PREC) GOTO 50
      WA(IP)=ZERO
      GOTO 55
   50 WA(IP)=TOP/H
   55 X(ITEMP)=HOLD
      I=I+1
      IF(I.LE.N) GOTO 40
      IF(K.LT.N) GOTO 60
      IP=IPART + N
      IF(ABS(WA(IP)).EQ.ZERO) GOTO 80
      X(ITEMP)=-E/WA(IP)+X(ITEMP)
      GOTO 100
C
C                  FIND PARTIAL DERIVATIVE OF LARGEST
C                  ABSOLUTE VALUE.
C
   60 KL=LKSUB+K
      LOOK=WA(KL)+PM1
      KMAX=LOOK
      IP=IPART+K
      DERMAX=ABS(WA(IP))
      KPLUS=K+1
      DO 65 I=KPLUS,N
         TEST=ABS(WA(IPART+I))
         IF(TEST.LE.DERMAX) GOTO 65
         DERMAX=TEST
         KMAX=I
   65 CONTINUE
      IF(LOOK.EQ.KMAX) GOTO 75
      LKMAX=LKSUB+KMAX
      WA(KL)=WA(LKMAX)
      WA(LKMAX)=LOOK
```

```
      IP=IPART+KMAX
      XTEMP=WA(IP)
      IPK=IPART+K
      WA(IP)=WA(IPK)
      WA(IPK)=XTEMP
      IF(K.LT.2) GOTO 75
      KMIN=K-1
      I1=0
      DO 70 I=1,KMIN
         L=((I1)*(N2-I))/2-1
         J=L+KMAX
         XTEMP=WA(J)
         JJ=L+K
         WA(J)=WA(JJ)
         WA(JJ)=XTEMP
         I1=I
   70 CONTINUE
   75 IF (ABS(WA(IPART+K)).NE.ZERO) GOTO 90
   80 IF (IERROR.EQ.1) GOTO 135
C
C                  IF THE MODIFIED JACOBIAN IS SINGULAR
C                  AT X, CHANGE THE COMPONENTS OF X AND
C                  PROCEED WITH THE ITERATIONS.  IF IT
C                  HAPPENS A SECOND TIME, TERMINATE.
C
      DO 85 I=1,N
         X(I)=0.9D0*X(I)+.12345D0
   85 CONTINUE
      IERROR=1
      GOTO 105
C
C                  SET UP COEFFECIENTS FOR THE KTH ROW
C                  OF TRIANGULAR LINEAR SYSTEM USED
C                  TO BACK-SOLVE FOR THE FIRST K X(I)
C                  VALUES.
C
   90 L=((K-1)*(N2-K))/2
      KN=L+N
      I1=L-1
      WA(KN)=ZERO
      IPK=IPART+K
      DO 95 J=KPLUS,N
         JSUB=WA(LKSUB+J)+PM1
         JJ=I1+J
         IPJ=IPART+J
         WA(JJ)=-WA(IPJ)/WA(IPK)
         WA(KN)=WA(KN)+WA(IPJ)*X(JSUB)
   95 CONTINUE
      LK=WA(LKSUB+K)+PM1
      WA(KN)=(WA(KN)-E)/WA(IPK) + X(LK)
      K = K+1
      IF  (K .LE. N) GO TO 10
C
C                  BACK SUBSTITUTE TO OBTAIN
C                  NEXT APPROXIMATION TO X
C
```

```
  100 IF (N.EQ.1) GOTO 105
      KMIN=N-1
      KK = 3
      GOTO 15
  105 IF (M.LE.1) GOTO 120
C
C                  TEST FOR CONVERGENCE..
C
      DO 110 I = 1,N
         IF(ABS(WA(ITMP+I)-X(I)) .GT. ABS(X(I))*RELCON) GO TO 115
  110 CONTINUE
      JTEST = JTEST+1
      IF (JTEST-3) 120,140,140
  115 JTEST=1
  120 DO 125 I=1,N
         WA(ITMP+I)=X(I)
  125 CONTINUE
  130 CONTINUE
      M=ITMAX
      IER=1
      GOTO 240
  135 IER=2
  140 FMAX=ZERO
      TEST=1.0E-15
      IF (N .GT. 1) GOTO 145
      CALL EVAL(X,F)
      WA(IPART+2)=F(1)
      FMAX = AMAX1(FMAX,ABS(WA(IPART+2)))
      GOTO 155
  145 CONTINUE
      CALL EVAL (X,F)
      DO 150 I = 1,N
         IP=IPART+I
         WA(IP)=F(I)
         FMAX=AMAX1(FMAX,ABS(WA(IP)))
  150 CONTINUE
C
C                  CHECK TO SEE IF SMALL COMPONENTS ARE
C                  ACTUALLY ZERO
C
  155 K=1
      DO 160 I=1,N
         WA(I)=X(I)
         IF(ABS(X(I)) .GT. P2) GOTO 160
         K=2
         WA(I)=ZERO
  160 CONTINUE
      IF (K .EQ. 1) GOTO 195
      KK = 1
      GOTO 205
  165 IF (FMAX.LT.TEST) GOTO 190
C
C            NOTE THAT SMALL COMPONENTS ARE SET
C            TO ZERO ONLY IF THE NORM OF THE
C            FUNCTION VECTOR IS REDUCED AS A
C            RESULT OF THIS PROCESS.
```

```
C
      DO 170 I=1,N
         X(I)=WA(I)
  170 CONTINUE
      IF  (N .GT. 1) GOTO 175
      WA(IPART+2)=WA(ITMP+2)
      GOTO 185
  175 DO 180 I=1,N
         WA(IPART+I) = WA(ITMP+I)
  180 CONTINUE
  185 FMAX=TEST
C
C             CHECK FOR INTEGER COMPONENTS
C
  190 K=1
  195 DO 200 I=1,N
         WA(I)=X(I)
         IF(ABS(X(I)) .LE. P2) GOTO 200
         L=X(I)+PT1
         J=X(I)-PT1
         IF (L .EQ. J) GOTO 200
         WA(I) =ISIGN(1,J)*MAX0(IABS(L),IABS(J))
         K=2
  200 CONTINUE
      IF (K .EQ. 1) GOTO 235
      KK = 2
  205 TEST=ZERO
      IF (N .GT. 1) GOTO 210
      CALL EVAL (WA,F)
      WA(ITMP+2)=F(1)
      TEST = AMAX1(TEST,ABS(WA(ITMP+2)))
      GOTO 220
  210 CONTINUE
      CALL EVAL (WA,F)
      DO 215 I=1,N
         IT=ITMP+I
         WA(IT)=F(I)
         TEST=AMAX1(TEST,ABS(WA(IT)))
  215 CONTINUE
  220 GOTO (165,225), KK
  225 IF (FMAX.LT.TEST) GOTO 235
C
C             NOTE THAT NEAR-INTEGER COMPONENTS
C             ARE SET TO BE INTEGERS ONLY IF THE
C             NORM OF THE FUNCTION VECTOR IS
C             REDUCED AS A RESULT OF THIS PROCESS.
C
      DO 230 I=1,N
         X(I)= WA(I)
  230 CONTINUE
C
C                    TEST FOR CONVERGENCE
C
```

```
  235 IF (FMAX.LE.EPS.OR.TEST.LE.EPS) GOTO 1005
      IF (IER.EQ.0) GOTO 1005
  240 IF (IER.EQ.2) WRITE (6,779)
      IF (IER.EQ.1) WRITE (6,778)
  779 FORMAT (/10X,'SINGULARITY OF THE JACOBIAN MATRIX',
     +        /10X,'HAS BEEN ENCOUNTED TWICE')
  778 FORMAT (/10X,'NO CONVERGENCE WITHIN THE ALLOWED',
     +        /10X,'NUMBER OF ITERATIONS')
 9005 RETURN
 1005 CONTINUE
      CALL EVAL (X,F)
      I=M1+1
      WRITE (6,1006) I
      WRITE (6,1007) (X(LL),LL=1,N)
      WRITE (6,1008) (F(LL),LL=1,N)
 1006 FORMAT (//10X,'SOLUTION FOUND AFTER ITERATION STEP',I3,/)
 1007 FORMAT (2X,'SOLUTION VECTOR',2X,8D14.7)
 1008 FORMAT (2X,'RESIDUALS',8X,8D14.7)
      RETURN
      END
```